A STRUCTURAL DESIGN METHOD FOR AN ARBITRARY TRANSVERSE SECTION OF A SHIP

by

THOMAS ARTHUR MARNANE
B. S. United States Naval Academy
(1957)

SUBMITTED IN PARTIAL FULFILLMENT
OF THE REQUIREMENTS FOR THE
DEGREES OF NAVAL ENGINEER
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Signature of Author . . . .
Department of Naval Architecture and Marine Engineering
May 22, 1964

Certified by . . . .
Thesis Supervisor

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Chairman, Departmental Committee
on Graduate Students

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A STRUCTURAL DESIGN METHOD
FOR AN ARBITRARY TRANSVERSE
SECTION OF A SHIP

by

Lt. Thomas Arthur Marnane, U.S.N.

Submitted to Department of Naval Architecture and Marine Engineering on 22 May 1964, in partial fulfillment of the requirements for the Master of Science Degree in Naval Architecture and Marine Engineering and the professional degree, Naval Engineer.

ABSTRACT

A procedure for carrying out an initial structural design of an arbitrary section of a ship is presented in this paper. The procedure is an extension of usual methods of midship section design. It includes the effects of shear loading. The method may be applied to either transversely or longitudinally framed ships. In both the worked example and the detailed procedure itself the calculations are as brief and the illustrations as simple as is considered possible. This permits the designer to achieve a reasonable first estimate in the shortest period of time. Several areas in which further research might be done are indicated.

Thesis Supervisor: J. Harvey Evans

Title: Professor of Naval Architecture

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INTRODUCTION

A great deal has been written in recent years concerning the rational design of the midship section of ships. Almost no attention has been paid to the design or verification of strength of other ships sections. Design criteria, derived from or based on classification society rules for the midship section, have been developed. Criteria based on theory and experience have also been generated. These criteria as applied to rational design methods have increased the speed and accuracy of the designer, at least in making first estimates. They have enabled him to expand his designs beyond the scope of the rules or previous experience with relative ease. They have not, however, provided him with a means for designing or verifying the design of a ships section upon which shearing forces act. This paper attempts to fill that gap.

In the example worked out in this paper and in the supplementary discussion in appendix (A) it is noted that the L/4 or quarterpoint section of the ship is the section considered. This in no way detracts from the generality of the method since this section was chosen merely for illustration purposes. Also, because midship structure is often arbitrarily extended to this section[1], it seemed to be a good point of departure for illustrative purposes.

The material used at any particular point in the ships structure may be chosen arbitrarily in the design procedure by the designer. No limitations are imposed on the user of this paper with relation to his choice of materials.

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1. American Bureau of Shipping, Rules for Building and Classing Steel Vessels, New York, 1962. Hereinafter referred to as "The Rules."

The author is fully aware that the method outlined herein has its limitations. In every instance where a distinct choice in method had to be made and whenever there was a question of addition or deletion of a design step or an additional check on stability or strength and if the choices were all within a reasonable range of accuracy for an initial design, the decision was always made on the side of the simplest, most direct approach. In addition, the entire approach is conservative. It is hoped that this procedure will encourage the practice of at least checking the scantlings of ship sections other than midships in preliminary design studies.

NOTATION

A = area, sq. ft.

a_1 = span of plating in line with direction of loading, in.

a_2 = larger panel dimension (shear and lateral type loading), in.

a = area of stiffener including plate, sq. in.

A_w = cross-sectional area of web material of a girder, in. x ft.

B = breadth of ship, ft.

b_1 = span of plating perpendicular to direction of loading, in.

b_2 = smaller panel dimension (shear and lateral type loading), in.

c = modified nondimensional column slenderness ratio.

C_B = block coefficient.

C = midships section coefficient.

C_p = prismatic coefficient.

D = depth of ship to strength deck, ft.

D_s = strut flexural rigidity, in. - lbs.

d = distance of member from baseline, ft.

E = Young's modulus, psi.

F = dimensionless coefficient relating column yield stresses.

F.S. = factor of safety.

H = full-load draft, ft.

h = head of water, ft.

I = moment of inertia of a structural cross-section, sq. in. x sq. ft.

i_o = moment of inertia of cross-section of a structural member about a transverse axis through the member, sq. in. x sq. ft.

K = nondimensional coefficient for bending moment.

K_b = nondimensional coefficient for buckling stress in flat plate in compression.

K_L = nondimensional coefficient for beading stress in plates under lateral load.

K_c = nondimensional coefficient for effective column length.

K_s = nondimensional coefficient for shear stress in flat plates.

K_v = nondimensional coefficient for shear load on a ship.

L = length of ship (waterline), ft.

L_w = length of wave, ft.

LBP = length between perpendiculars, ft.

M = bending moment, ft. - tons.

Q = first moment of a cross-sectional area about neutral axis, sq. in. x ft.

r = radius of gyration referred to buckling axis, in.

s = frame spacing.

t = plating thickness, in.

v = shearing force, tons.

y = distance of structural element from neutral axis of bending, ft.

Y_B = distance from keel to neutral axis of bending, ft.

Z = section modulus of a structural cross-section, sq. in.x ft.

Δ = full-load displacement, tons.

σ_1 = ship bending stress, psi.

σ_2 = girder bending stress, psi.

σ_3 = plate bending stress, psi.

σ = bending or direct stress, psi.

σ_{cr} = critical direct stress, psi.

σ_{cR} = applied stress causing yielding or buckling of column, psi.

σ_L = principle stress (longitudinally framed ship), psi.

σ_t = principle stress (transversely framed ship), psi.

σ_p = principle stress, psi.

σ_y = yield strength in tension or compression, psi.

σ_x = direct stress in longitudinal direction, psi.

σ_y = direct stress in vertical or transverse direction, psi.

σ_1' = maximum safe bending stress.

m = Poisson's ratio.

ρ = density of water, lbs. x cu. ft.

τ = shearing stress, psi.

τ_{xy} = shearing stress along one of coordinate axes, psi.

τ_{cr} = critical shearing stress, psi.

= pounds.

Subscripts

B = bottom plating

D = deck plating

L = longitudinal framing

max. = a maximum value

min. = a minimum value

m = main longitudinal

p = plating

t = transverse framing

T = total

s = stiffener

sp = side plating

w = working

DESIGN CONSIDERATIONS

Structure and Stress System

Consider the L/4 section of a ship. At various positions in this section shear and bending stresses are acting. At the neutral axis the stress is zero (with the vessel upright, of course). At the center line the shearing stress is zero. At the bilge strake both shear and bending stresses act. Thus in this structure various stresses act together or singly in various portions of the structure. "Various stresses" and "various portions" are not particularly concise descriptions, however, so the following defined system, initially described by St. Denis[2] and Evans[3] is set forth.

All structural elements or assemblies are divided into three types which are defined as follows:

1. Primary structure - Structure of quasi-infinite rigidity in the plane of loading. This type of structure includes shell, bulkheads, decks, and inner bottom loaded in their planes.

2. Secondary structure - Structure of finite rigidity or flexibility in the plane of loading. This structure includes only stiffened structure, that is, shell, bulkheads, decks, double bottoms, frames, floors, webs, and longitudinals which are loaded normally.

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2. St. Denis, M., On The Structural Design of the Midship Section, p. 10.

3. Evans, J. H., A Structural Analysis and Design Integration with Application to the Midship Section Characteristics of Transversly Framed Ships. Trans. SNAME, vol. 66, 1958, p. 244.

3. Tertiary structure - Structure of small rigidity (extreme flexibility) in the plane of loading. All unstiffened plating loaded normally is included.

Stresses which correspond to these types of structure are defined as follows:

1. σ_1 - Primary Stress. That stress which is caused by the over-all ship bending moment. The ship structure acts as a beam upon which buoyancy and weight differences act to create resisting bending moments. In an upright condition this stress is generally assumed constant across the deck and bottom plating cross section and is proportional to the distance from the neutral axis.

2. σ_2 - Secondary stress. That stress arising from the application of a normal loading to a plating-stiffener combination. The stress in a cross section of the plating is, due to shear lag, a maximum in way of the stiffener and diminishes with increasing distance from it.

3. σ_3 - Tertiary stress. This stress, also called plate bending stress, occurs when a simple plate panel, supported on its four edges, is subjected to lateral loads.

4. τ - Shear stress. That stress arising from vertical shearing forces caused by ship loading and acting on a transverse section of the ship.

5. Torsional stresses are not considered but may be present.

Ship Loading

Since this paper is dealing with a general method of making at least a reasonable first estimate or check of the scantlings of an arbitrary ship section some assumptions concerning loading must be made. These

assumptions must at best be approximate but with some explanation of the method of approximation and some understanding of range of variation they can be used intelligently. Bending moment and shear will be considered separately and then together.

In generating the primary stress Evans[4] suggests the "time honored" $\Delta L/35$[5] as a reasonable estimate of maximum bending moment.

Arnott[6] suggests setting

$$\Delta = \frac{LBHC_B}{35} \text{ tons}$$

and then

$$M = \frac{L^2BHC_B}{35 \times 35} \text{ ft. - tons.} \qquad (1)$$

He further states that no loss in accuracy is implied (at least for merchant ships) by substituting 0.75 for C_B. Bending moment calculations on a standard trochoidal wave are then carried out for a number of ocean-going passenger vessels with machinery amidships and compared with those derived from Eq. (1). In most cases the Eq. (1) value is the largest (most conservative) and in the cases where the calculated value is largest in only one case is the difference significant. When the

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4. Evans, op. cit., p. 249.

5. Standard symbols have been used whenever possible. A complete list of symbols and meanings is given at the beginning of this paper.

6. Arnott, D., ed., Design and Construction of Steel Merchant Ships, p. 97 - 100.

machinery is aft Arnott concludes that an increase of ten to fifteen percent in the Eq. (1) value is justified for design purposes. He bases this conclusion on tanker data in a paper by McDonald and MacNaught[7].

This author intends to use the general form

$$M_{max} = \frac{\Delta L}{K} \tag{2}$$

of which the formulae in the preceding discussion can be seen to be specific cases. Here K is a nondemensional coefficient which depends on ship type. It has often been pointed out that by finding actual ship bending moments and working backwards values of K can be found. Some typical values are:[8]

1. British Warships (Larger Than Destroyer)
   a. Hogging Condition $19.4 < K < 43.9$
   b. Sagging Condition $23.8 < K < 50.9$
2. British Destroyers
   a. Hogging Condition $21.0 < K < 22.7$
   b. Sagging Condition $24.4 < K < 29.0$
3. Light Cruisers
   a. Hogging Condition $26.4 < K < 30.3$
   b. Sagging Condition $25.1 < K < 29.0$

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7. McDonald and MacNaught, Investigation of Cargo Distribution in Tank Vessels, Trans. SNAME, Vol. 57, 1949, p. 483.

8. Caldwell, J. B., Naval Structural Theory, unpublished lecture notes, Naval Architecture Department, M.I.T., 1963.

4. Aircraft Carriers

a. Hogging Condition $20.7 < K < 27.9$

b. Sagging Condition $30.7 < K < 35.8$

These figures are quoted here in order to demonstrate the range of values over which K may roam for naval ships. The figure of K equal to 35 for merchants has been previously quoted in this paper. Other values of K, along with the above, are plotted in Fig. (I) which will be discussed later.

Study of the naval ship K values quoted indicates the liklihood of a larger hogging than sagging bending moment. This indicates that if the machinery is not always exactly amidships, at least it is not disposed near the extremes either. Thus for the range of naval ships the assumption that the hogging bending moment is the greatest is reasonable. The sagging condition should, time permitting, be checked, however, especially if, as for the cruisers, there is no prior data clearly indicating which condition is the worst. This may be further illustrated if one considers that the difference between $\frac{\Delta L}{35}$ (machinery amidships) and $\frac{\Delta L}{26}$ (machinery aft) taken as average values is about thirty-five percent for the maximum hogging moment.

A parabolic distribution of moment over the ship length is considered reasonable by this author and will be assumed without further justification.

Evans[9] suggests

$$V_{max} = \Delta/8.$$

---

9. Evans, op. cit., p. 272.

This author feels that this formula is of the proper form, i.e.

$$V_{max} = \frac{\Delta}{K_v} \tag{3}$$

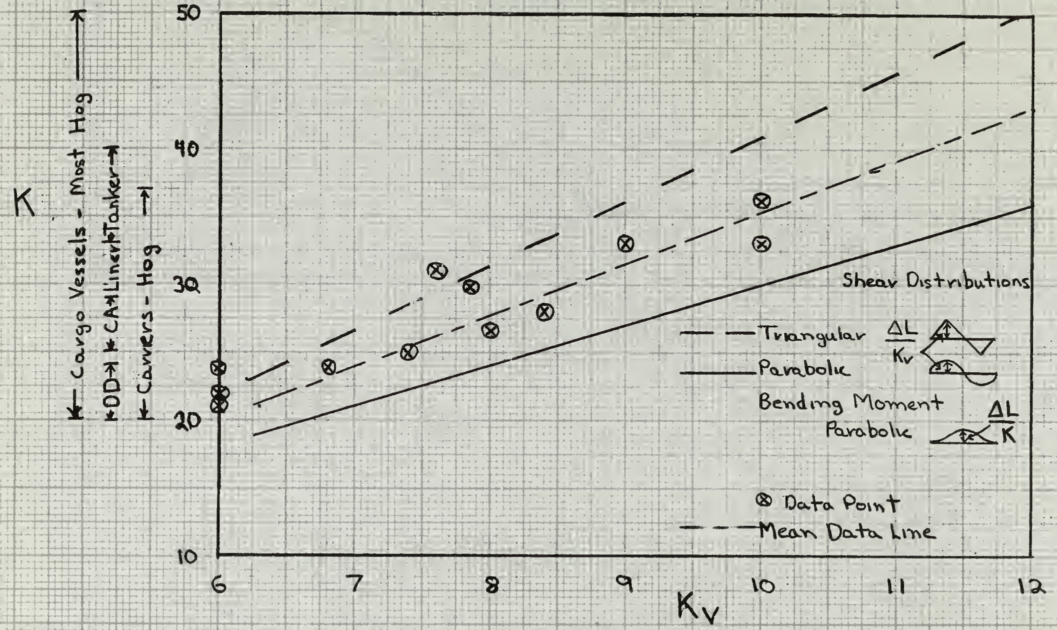
where K_v is a nondimensional constant related to the ship considered. It would seem further that the shear loading is subject to some of the same limitations and criteria as the bending moment. The area under a ships shear curve from either extremity up to a given station should equal the bending moment at that point, that is

$$\int_0^x V\, dx = M \tag{4}$$

If, as may be reasonable in view of the previously assumed parabolic moment distribution, a triangular distribution of shear is assumed, it is seen that Eq. (4) yields a maximum bending moment of $\Delta L/32$ for a shear of $\Delta/8$ and for a shear of $\Delta/9$ the maximum moment is $\Delta L/36$. A parabolic distribution of shear for the same shear values yields moments of $\Delta L/24$ and $\Delta L/27$ respectively. These values are obviously quite different and it appears that, as for the bending moment, no set value of shear is entirely adequate for an initial estimate. For this reason Fig. (I) has been developed.

In Fig. (I) values of maximum bending moment have been plotted for both triangular and parabolic distributions of shear. The range of values of K as taken from actual ship measurements have also been plotted. Finally for the cases where both K and K_v values have been measured these values are plotted. Based on this admittedly scanty

Bending Moment Constant vs. Shear Load Constant



Data References: Rossel, H. and Chapman, L. Principles of Naval Architecture, Vol. I, SNAME, New York, 1939.
U.S. Maritime Commission, Structural Tests on the S.S. Fort Mifflin, Shiloh, and Ventura Hills.
Bates, J. and Wanless, I. Aspects of Large Passenger Liner Design, Trans. SNAME, Vol. 54, 1946.
Caldwell, J.B. Warship Structural Theory, Unpublished Lecture Notes, Nav. Arch. Dept., MIT, 1963.
Vasta, J., Lessons Learned From Full Scale Structural Tests, Trans. SNAME, Vol. 66, P. 170, 1958.

FIGURE I

Bending Moment and Shear Load Along Ships Length vs. Percent of Maximum Values

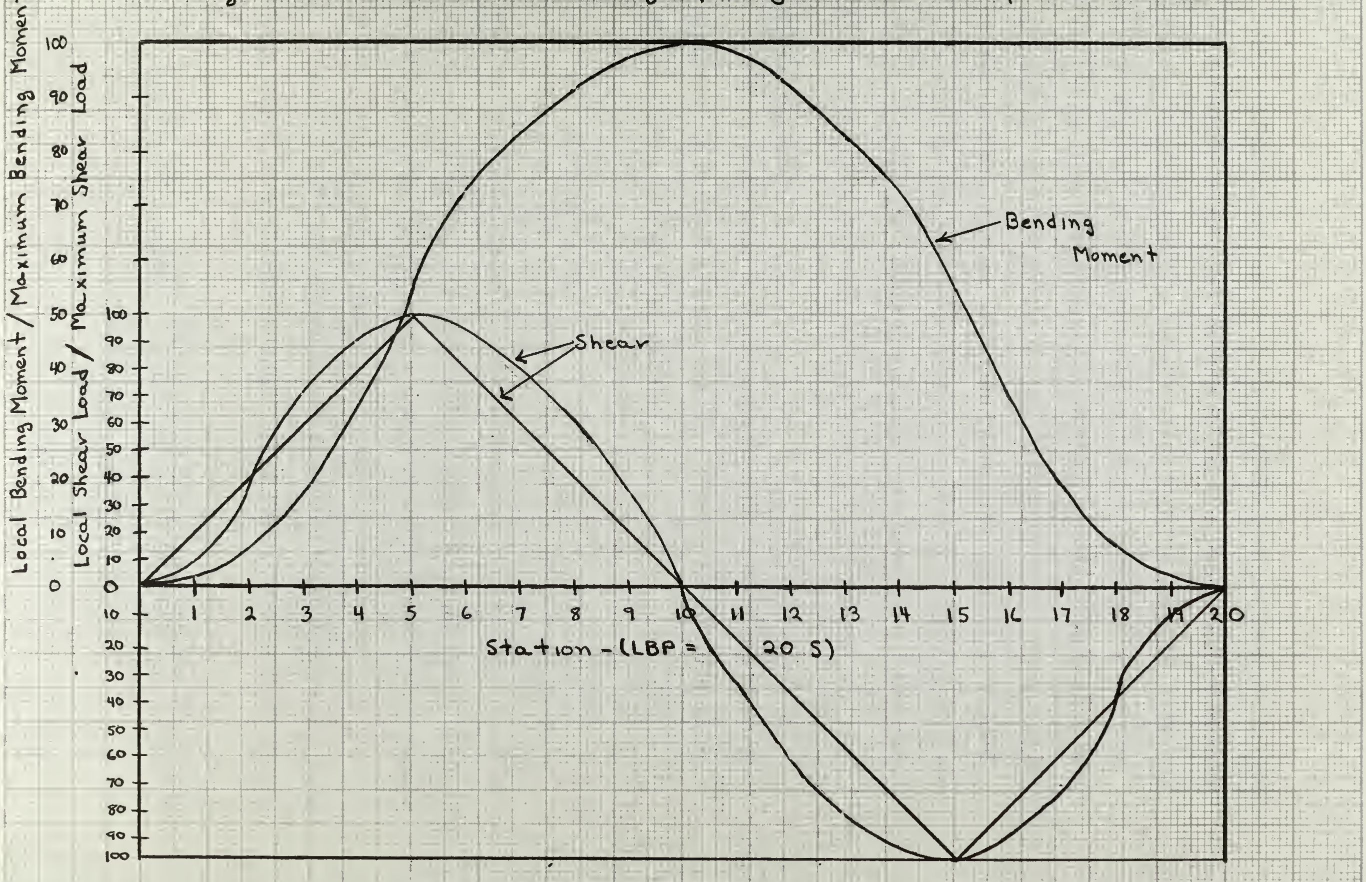


FIGURE II

data it is nonetheless felt that a line midway between those for parabolic and triangular shear distributions might have been expected based on observation of shear force distributions on typical ships. Thus, for a first estimate of maximum shearing force, assumed to occur at the L/4 section, and of maximum bending moment, assumed to occur at midships, the center line of Fig. (I) is used in this paper. In order to facilitate use of these maximum values Fig. (II) is also plotted. This plot gives percent of maximum values of shear and bending along the ships length for the loading distributions previously discussed. Only one bending moment curve is plotted for simplicity.

Stress Criteria

If now, in addition to the bending moment distribution already discussed, a reliable criteria for σ_1, bending stress, can be set down, the required section modulus

$$Z = \frac{M}{\sigma_1} \tag{5}$$

may be determined readily. This author will limit his discussion to existing criteria for mild steel and will assume that for other materials or different steel types the safe design stress is directly proportional to that for mild steel in the ratio of yield stress values.

Evans[10] gives two equations and compares these with the Load Line Regulations[11]. He later concludes that the Rules can be reasonably

10. Abell, W. S., Some Questions in Connection with the Work of the Loadline Committee, Trans. INA, vol. 58, 1916, pp. 16-36.

11. United States Coast Guard, Load Line Regulations.

approximated by

$$\nabla_1 = 5\,(1 + \frac{L}{1000})\,2240 \tag{6a}$$

and

$$\nabla_1 = 2240\ \sqrt[3]{L}\,. \tag{6b}$$

Arnott[12] proposes that

$$\nabla_{1_{max}} = 1.19\ \sqrt[3]{L}\ ^{13} \tag{7}$$

for ships in excess of 250 feet long and he presents a plot, Fig. (III), illustrating his method of arriving at the 1.19 factor. By comparison of the two curves the value of maximum stress for any length ship may be found. It is this author's opinion that Eq. (7) together with Fig. (III) is at least as good as either of Eqs. (5) or (6). This is based on the fact that as the ship increases in size there is a corresponding increase in design stress, at least up to 250 feet. When one considers the fact that a fairly standard corrosion allowance is about one-eighth inch for both small and large ships, the increased stress for the smaller ships, in which the same amount of corrosion reduces the over-all strength by a greater percentage than for a large, thicker shelled ship, is justified. Equation (7) is used in this paper.

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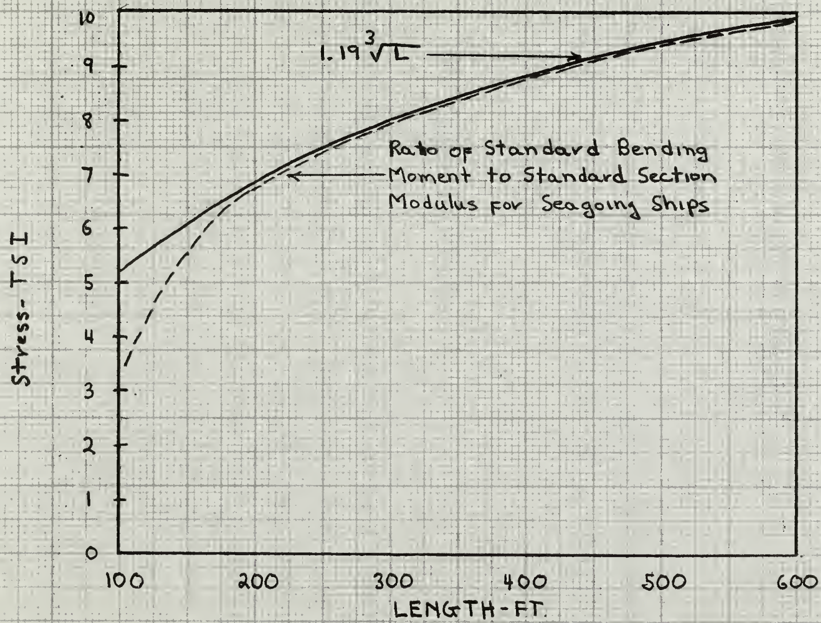
12. Arnott, op. cit. p. 103.

13. Johnson, A. J. and Larkin, E., Stresses in Ships in Service, Figure 9, also tends to verify this equation.

FIGURE III



Ref: Arnott, D., *Design and Construction of Steel Merchant Ships*, p. 103, SNAME, New York, 1955.

Constant values of secondary bending stresses will be used in this paper. In the longitudinal direction

$$\sigma_2 = 2000 \text{ psi}$$

will be used and in the transverse direction

$$\sigma_2 = 3000 \text{ psi}$$

will be used. These are the same values as used by St. Denis[14].

The expression for plate bending stress, σ_3, under a uniform hydrostatic load is given by[15]

$$\sigma_3 = \frac{1}{288} K_L \quad h \left(\frac{b_2}{t} \right)^2 . \tag{8}$$

K_L is a modifying coefficient varying with the plate edge concerned and may be found by using Fig. (IV). Hydrostatic loading on the ships plating is not, of course, uniform. If, however, the head of water at any given plate is taken to be either that existing at thirty degrees of roll in still water or as that at twice the full load draft, then the use of this equation based on the location of the middle or lower edge of a plate will be at worst conservative.

In the consideration of shear stress criteria consider the equation

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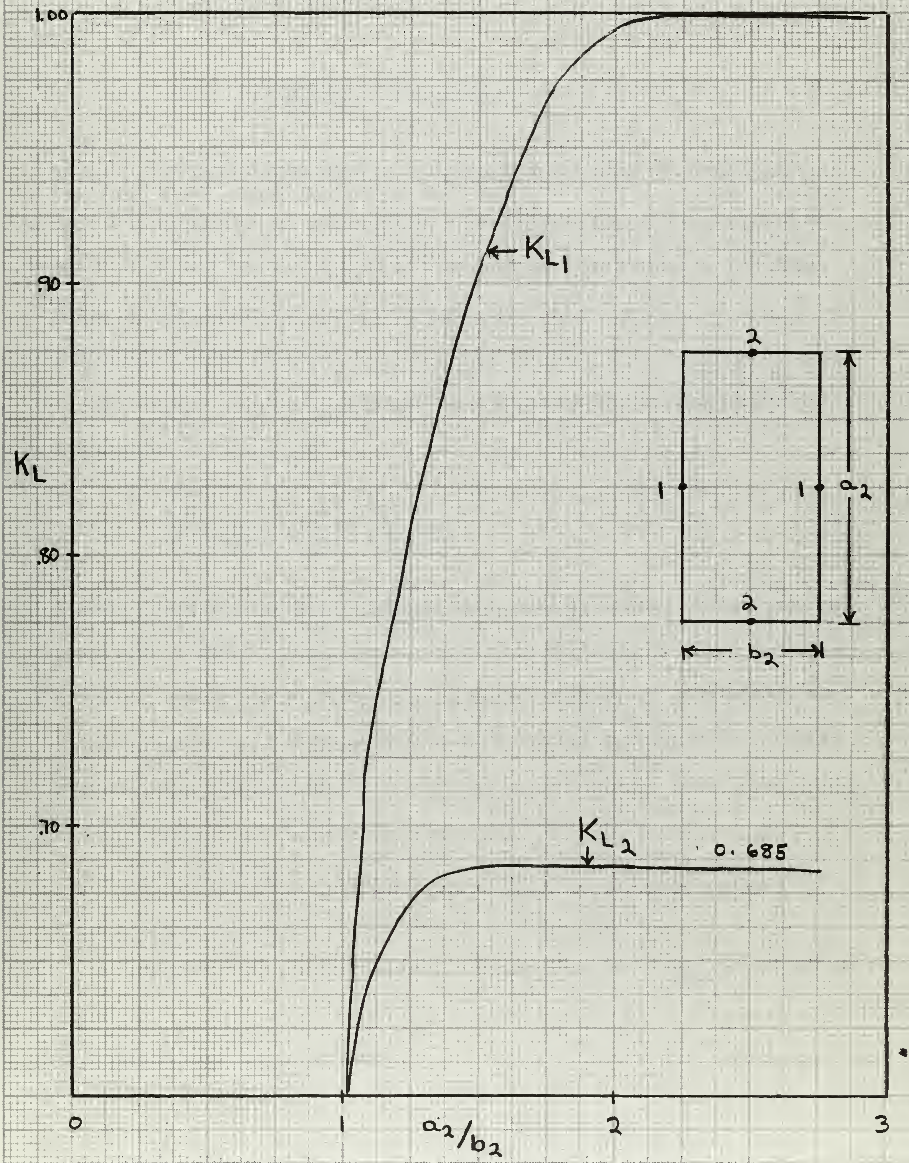
14. St. Denis, op. cit., pp. 50-51.

15. Evans, op. cit. p. 251.

FIGURE IV

K_L Constants For Equation (8)



Ref: Evans, Op. Cit., p. 250.

$$\tau_{max} = \frac{2240\, V\, Q_{max}}{12\, t\, I} \tag{9}$$

$$= \frac{2240\, V}{12\, t\, A_w} \times \text{constant} .$$

If the beam shear approximation is made for a ship then the constant above is a function of the relative area of material in flanges and in the web of an I beam. A plot of a maximum shear stress, average shear stress ratio versus the flange, web ratio for several beams indicates a value of about 1.1[16] for the constant for the range of ships.

If then for a ship A_w is taken (as a first approximation) equal to 2 t D then

$$\tau_{max} = 103.0 \; \frac{V}{t\, D} \; \text{psi} \tag{10}$$

where t is essentially an average side shell thickness in this development and in Eq. (10) is necessarily that at the neutral axis. Consider, however, some other point, say the sheer strake. It is apparent that if the thickness at the sheer strake is the same as that at the neutral axis then the shear stress at the sheer strake will be some percentage of the maximum as determined from Fig. (V). If, however, the thickness varies, as it usually does, then this procedure will have to be slightly modified. This can be done as follows:

1. Assume the shear stress at neutral axis is $\tau_{max} = \frac{\text{constant}}{t}$.

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16. Evans, op. cit., p. 272

2. For constant thickness the shear stress at some other point on the side will be

$$\tau = \frac{(\text{constant})\ (\text{percentage} \leq 100 \text{ percent})}{t}$$

3. If the thickness varies then

$$\tau = \frac{(\text{constant})\ (\text{percentage} \leq 100 \text{ percent})}{\cancel{t}} \quad \frac{\cancel{t}}{t\ (\text{local})}$$

Thus it is seen that t in Eq. (10) can be treated as local plating thickness rather than that at neutral axis.

Even when the maximum shearing stress has been determined the distribution of shearing stress around the periphery of a particular cross-section remains to be ascertained. When nothing is known of the structure this is not an insignificant problem. No hard and fast rules or equations seem to be available for solution of this problem. Some shear measurements are available but not in sufficient number and detail.

The author, therefore, has devised a simple approximation, the results of which are shown in Fig. (V). This plot was arrived at using the following assumptions.

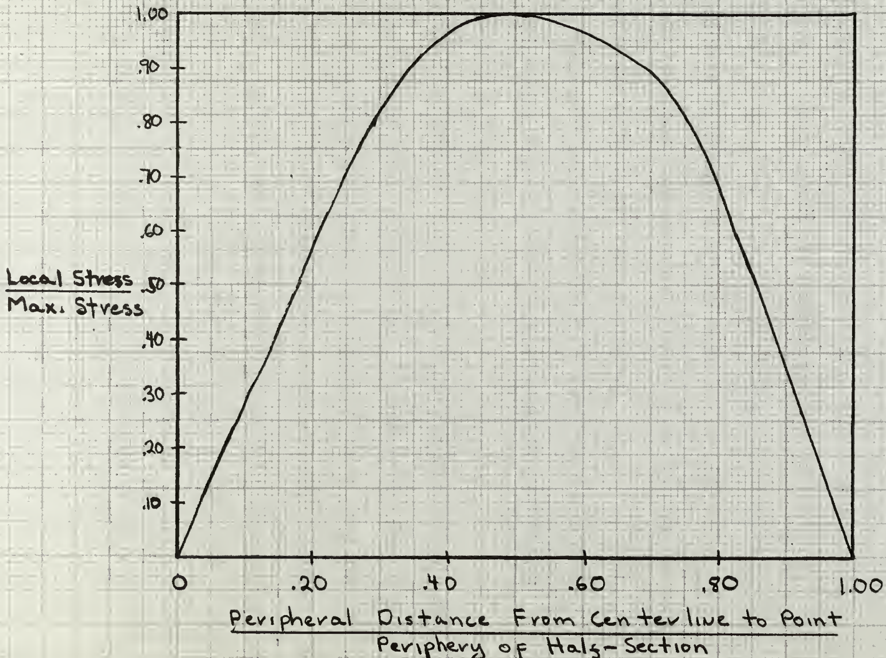
1. The neutral axis for the range of ships lies, in general, between .4D and .5D. Therefore, .45D is a reasonable approximation for its location.

2. The shape of a ships section may be assumed to lie somewhere between a rectangle and a triangle in broad outline.

3. B $\approx$ 1.3D for the range of ships.

FIGURE V

Peripheral Distribution of Shear Stress



Note: Read Discussion Before Using This Plot. Plot Based on Comparison of Various Cross-Sectional Shapes with

$B \approx 1.3D$

$.4D \leq Y_B \leq .5D$

4. The section has no decks which need to be treated as contributing to longitudinal strength. Using Fig. (V) then, one need only enter with the point at which the shearing stress on the section is desired and read off this ratio of the local stress to the maximum stress. If there are decks present which are considered in the longitudinal strength the shearing stress is modified as follows:

1. All shearing stress values determined with no decks should be reduced at points above the deck and along the side by fifteen percent if the deck is above the neutral axis. This reduction is cumulative for each deck above the neutral axis.

2. Similarly, reduce the values of shearing stress down and along the side for each deck below the neutral axis.

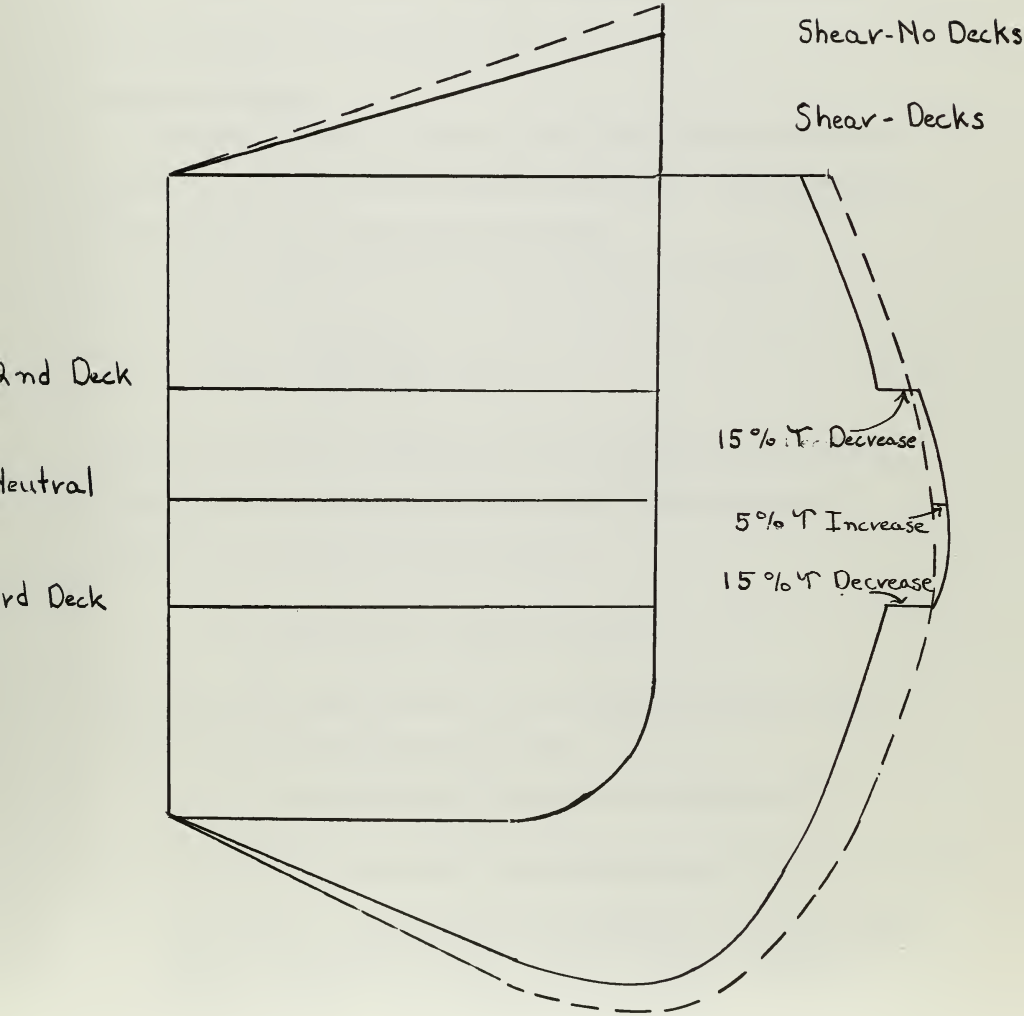
3. Increase the shearing stress at neutral axis by five percent over that previously determined with no decks.

Thus for a ship with strength decks both above and below the neutral axis one might have a distribution of shearing stress as shown qualitatively in Fig. (VI).

At this point it may seem to the reader that the preceding method is so approximate as to be of little value. If, however, it is kept in mind that this is an approximation which need be made only once in a design cycle, that the values obtained for a typical section location in appendix (A) do not exceed sixty percent of the total longitudinal stresses, and that the yield stress in shear is often taken as about $.6\sigma_y$, then the approximations are not completely unreasonable. The peripheral distributions are also reasonable. Based on observation of any

FIGURE VI

Shear Stress Distribution
Around Typical Section



typical sections it may be seen that the material contributed by an additional deck and its stiffeners decreases $\frac{Q}{I}$ in Eq. (9) in such a manner as to give the fifteen percent and five percent figures previously quoted and illustrated in Fig. (VI).

Instability Criteria

There are a number of possible criteria and equations which may be used for critical buckling stress of plate panels. Without attempting to justify them this author uses the following:

1. For a longitudinally framed ship and due to Bryan[17]

$$\sigma_{cr} = \frac{K_b \pi^2 E}{12(1 - m^2)} (t/b_1)^2 . \qquad (11)$$

If it is assumed that all edges of a plate are pinned but free to rotate then K_b is approximately equal to four and for widely spaced deck beams with longitudinals remaining straight

$$\sigma_{cr} \text{ (mild steel)} = 10830 \times 10^4 \left(\frac{t}{b_1}\right)^2 .$$

For values of K_b other than four, i.e., for other end conditions and closely spaced deck frames see Fig. (VII).

2. For a transversly framed ship and due to Montgomerie[18]

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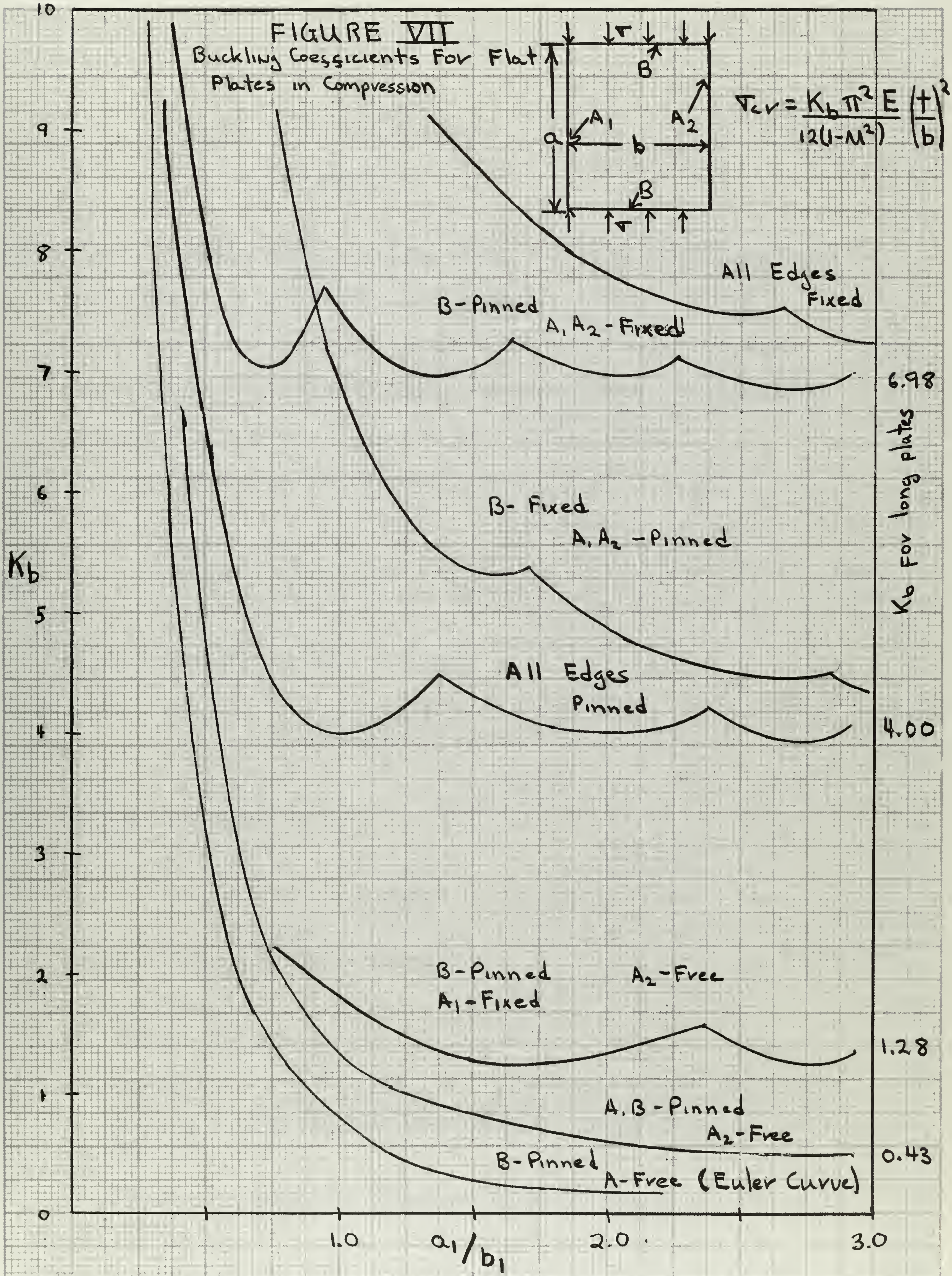
17. Bryan, G. H., Proceedings London Mathematical Society, v.22, p. 54.

18. Montgomerie, J., Experiments on the Compression of Samples of Deck Plating and the Application of the Results to Determine a Safe Limiting Thickness for Weather Decks in Certain Conditions of Loading, Journal of the Society of Naval Architects of Japan, vol. 54, 1934, pp. 121-163 (English translation).



Ref. Young. Ship Plating Subject to Loads Both Normal to and on the Plane of the Plate, TRINA, Vol. 99, p. 578, 1959.

$$\sigma_{cr} = \frac{40,300}{1 + \frac{1}{950}\left(\frac{a_1}{t}\right)^{1.75}} \qquad (12)$$

There are restrictions on the use of Eq. (12) as pointed out by Evans[19] and others but it is still the best available.

Sezawa[20] offers similar equations for both a wide plate with clamped edges and one with simply supported edges. Montgomeries formula falls in between as can be seen in Fig. (VIII).

For critical shearing stress this paper uses[21]

$$\tau_{cr} = \frac{K_s \pi^2 E}{12(1 - m^2)} \left(\frac{t}{b_2}\right)^2 \qquad (13)$$

in which for simply supported edges

$$K_s = 5.35 + 4 \left(\frac{b_2}{a_2}\right)^2$$

and for clamped edges

$$K_s = 8.98 + 5.6 \left(\frac{b_2}{a_2}\right)^2 .$$

Torsional shearing will be neglected.

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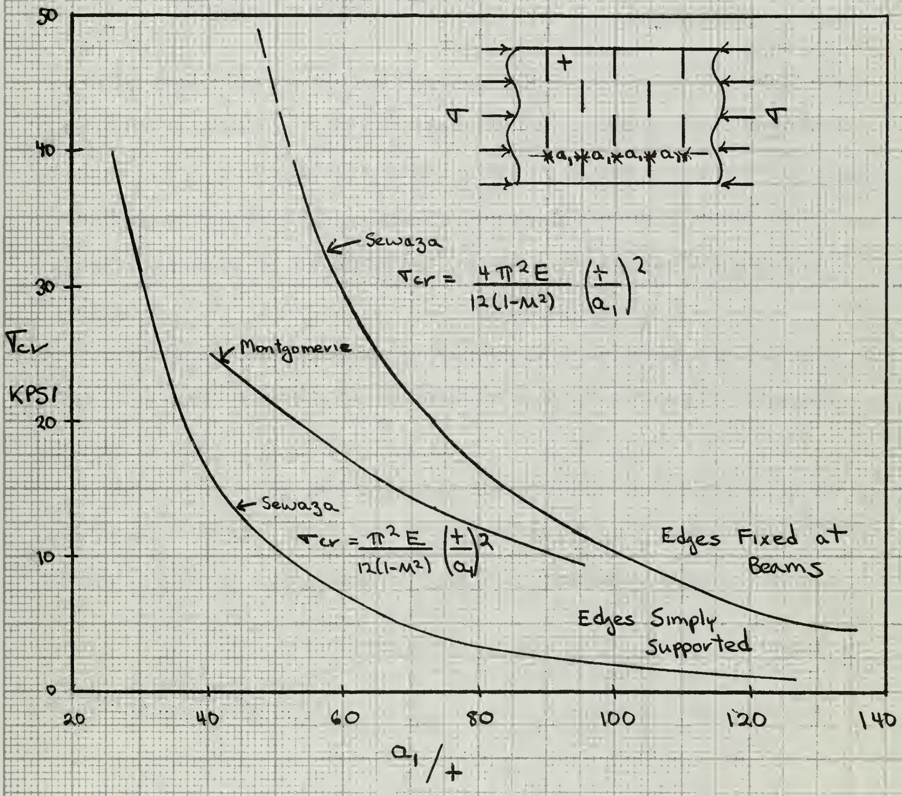
19. Evans, op. cit., p. 247 .

20. Sezawa, K., and Watanabe, W., Buckling of a Rectangular Plate with Four Clamped Edges Re-examined with Improved Theory, Reports, Tokyo Imperial University, Aero Research Inst., Vol. 11, No. 143, 1936.

21. Bleich, F., Buckling Strength of Metal Structures, p. 390-395.

FIGURE VIII

Buckling Stresses For Steel Deck Plating
Transverse Framing



In the case of combined shearing and bending stresses the interaction formula[22]

$$\left(\frac{\tau_w}{\tau_{cr}}\right)^2 + \left(\frac{\sigma_w}{\sigma_{cr}}\right) \leq 1 \qquad (14)$$

will be used. Satisfaction of this equation indicates no buckling. This equation is valid for $0 \leq \frac{\sigma_w}{\sigma_{cr}} \leq 1$ and for $\frac{a_1}{b_1} \leq 1$ and in the elastic range. τ_w will be local average shearing stress at the point or panel considered and σ_w will in general delineate the longitudinal compressive stress (or tensile stress treated as compressive) which is acting.

Hydrostatic lateral loadings will not be considered significant in buckling considerations for outer shell plating or inner bottoms and decks.[23]

Factors of safety which may be used with the above are up to the designer. The following are offered, however, for general use.[24]

1. $\frac{\tau_{cr}}{\tau} \geq 1.5$ where τ_{cr} is found from Eq. (13).

2. $\frac{\sigma_{cr}}{\sigma_w} \geq 1.5$ for plate buckling where σ_{cr} is determined from Eqs. (11) or (12).

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22. Bleich, F., op. cit., p. 405.

23. Bleich, F., op. cit., p. 498.

24. Institution of Structural Engineers, Report on Structural Safety, The Structural Engineer, vol. 34, no. 5, p. 141, May 1955.

The stability of the longitudinal strength and stiffening members must be considered. Figure (IX) is used for this purpose in this paper and its use is self-explanatory. A factor of safety of

$$\frac{\sigma_{cr}}{\sigma_w} \geq 1.75$$

is reasonable for this application.

Finally, in order that the critical buckling stress equations for the plating may be considered valid, the longitudinals must remain straight. If

$$\frac{EI}{bD_s} \geq 21.5 \left(\frac{a_1}{b_1}\right)^2 - 7.5 \quad (15)$$

[25]

then the longitudinal does indeed remain straight.

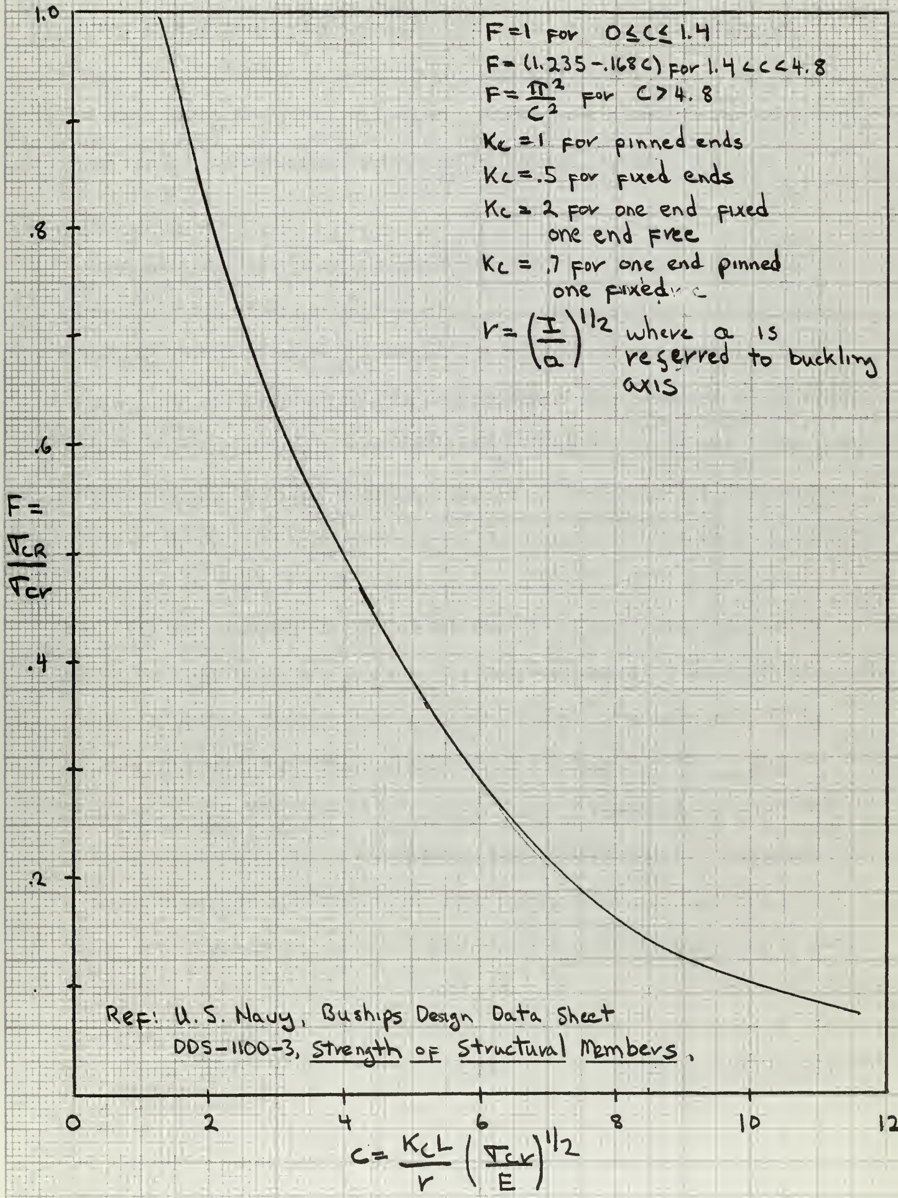
Superposition of Stresses

If one considers a stressed panel of plating at all of its points, that is, along the edges (on both the tensile and compressive sides if the plate is subjected to bending stresses) and in the center (again on both sides) it will be seen that the maximum existing stresses in any direction in the plane of the plate have the same sign (compression or tension) at at least one point. Thus the maximum stress conditions in any plate may be determined by simply adding the maximum stresses existing in the two coordinate directions (excluding, however, shearing stresses). This fact will be used in this paper. If all three stresses, σ_1, σ_2, and σ_3 exist in a plate one need simply add their absolute

25. Harlander, L. A., Optimum Plate-Stiffener Arrangement for Various Types of Loading, Journal of Ship Research, vol. 4, no. 1, 1960 p. 49.

FIGURE IX

Buckling Strength of Longitudinal Stiffeners



values and then using a Mohr's circle[26] analysis include the shearing stress to determine the principle stresses in the transverse and longitudinal or vertical and longitudinal directions.

This same principal of addition of stresses holds for any structural member if consideration is limited to a single plane.

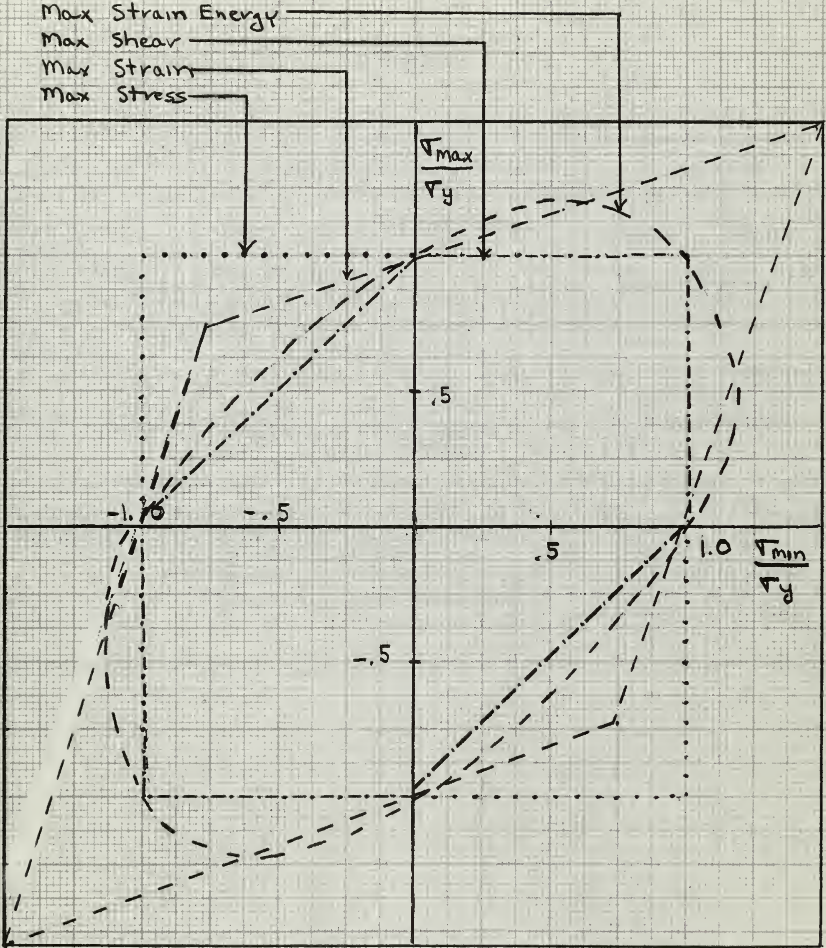
Theories of Failure

At different points in a ships structure various combinations of previously discussed stresses exist. As has also been noted, these stresses can be combined, utilizing a Mohr's circle approach, into maximum and minimum principle stresses. The question as to what values of these stresses will initiate yielding is of interest. The answer depends on the yielding criteria adopted. There are five well-known yield criteria. The four principle ones are considered here. These are listed in Table 1 and plotted in Fig. (X). Careful scrutiny of Fig. (X) will indicate to the reader that if one knew the quadrant in which the maximum and minimum stress values lay it would be a simple matter to decide which criteria he would use. For example, if the results of a Mohr's circle analysis yielded a negative σ_{min} and a positive σ_{max} then the result would lie in the second quadrant and any one of three criteria would probably be reasonable. Investigation of various stress conditions, as done in appendix (A), indicates that there is no rule or set of rules which will enable the designer to decide in advance which criteria he should use. He must run through a complete

26. Timoshenko, S. and MacCullough, G. H., Elements of Strength of Materials, pp. 64-70.

FIGURE X

Theories of Failure



Ref: Marin, J., Mechanical Properties of Materials and Design, p.49.

TABLE I

Criterion[27]	Yielding Occurs When	Equation
Shear Strain Energy (Mises-Hencky)	Shear energy per unit volume = shear energy in simple tension specimen	$\sigma_{pmax}^2 + \sigma_{pmin}^2 - \sigma_{pmax}\sigma_{pmin} = \sigma_y^2$
Maximum Stress (Rankine)	Maximum principle stress = yield stress in tension	$\sigma_{pmax} = \sigma_y$
Maximum Shear (Guest-Tresca)	Maximum shear stress = maximum shear stress at yield in simple tension	$\sigma_{pmax} - \sigma_{pmin} = \sigma_y$
Maximum Strain (St. Venant)	Maximum unit strain = unit strain for yielding in simple tension or minimum unit strain in compression = unit strain in simple compression	$\sigma_{pmax} - m(\sigma_{pmed} + \sigma_{pmin}) = \sigma_y$ or for a thin plate $\sigma_{pmax} - m\sigma_{pmin} = \sigma_y$

27. More complete explanations of these are found in
    a. Timoshenko and MacCullough, op. cit., pp. 374-378.
    b. Timoshenko, Strength of Materials, part II, pp. 473-482.

Mohr's circle analysis first. It was noted, however, in view of the discussion on superposition of stresses in which it was assumed that all stresses could be taken with like sign, and after plotting many Mohr's circles, that the results were in the first and third quadrants <u>most of the time</u>. It was further noted that the two criteria which were most consistently in agreement and still fairly conservative were the Mises-Hencky and Guest-Tresca. Since, historically, the Mises-Hencky criteria has been preferred and since application of either the Mises-Hencky or Guest-Tresca criteria is equally laborious (as will be seen later in the design method and example), the Mises-Hencky criteria is chosen for application in this paper.

<u>Degree of Fixity</u>

In this paper longitudinals and side plating are considered fixed against lateral pressure and pinned in longitudinal bending. These assumptions are made on the basis that for complete or full fixity to exist:

1. There must exist adequate bracketing and the stiffener ends must be welded to stiff structure whose i_o/length ratio is large compared to the stiffener.

2. The stiffener must be continuous over a support and symmetrical loading must exist on each side of the support. In this paper it is also considered that

3. There is no rotation of the tangent to the elastic curve (as between the loaded and unloaded conditions).

In the case of lateral loading condition three is exactly fulfilled. Thus both one and two must be fulfilled. Thus the stiffeners and edges of plating are considered fixed. These conditions are not fulfilled exactly in longitudinal bending. Thus the ends and edges are treated as pinned.

Transverse and Longitudinal Framing

This author considers that it is a fair statement to make that the transversely framed ship will differ in the method of design from a longitudinal framed ship in essentially only two aspects. That is, in the amount of longitudinally continuous material which is considered in longitudinal strength calculations and in the plating aspect ratios which are used. Thus the designer need only keep in mind these two general conditions when carrying out his design and the method outlined in this paper remains valid for initial design involving any type of framing.

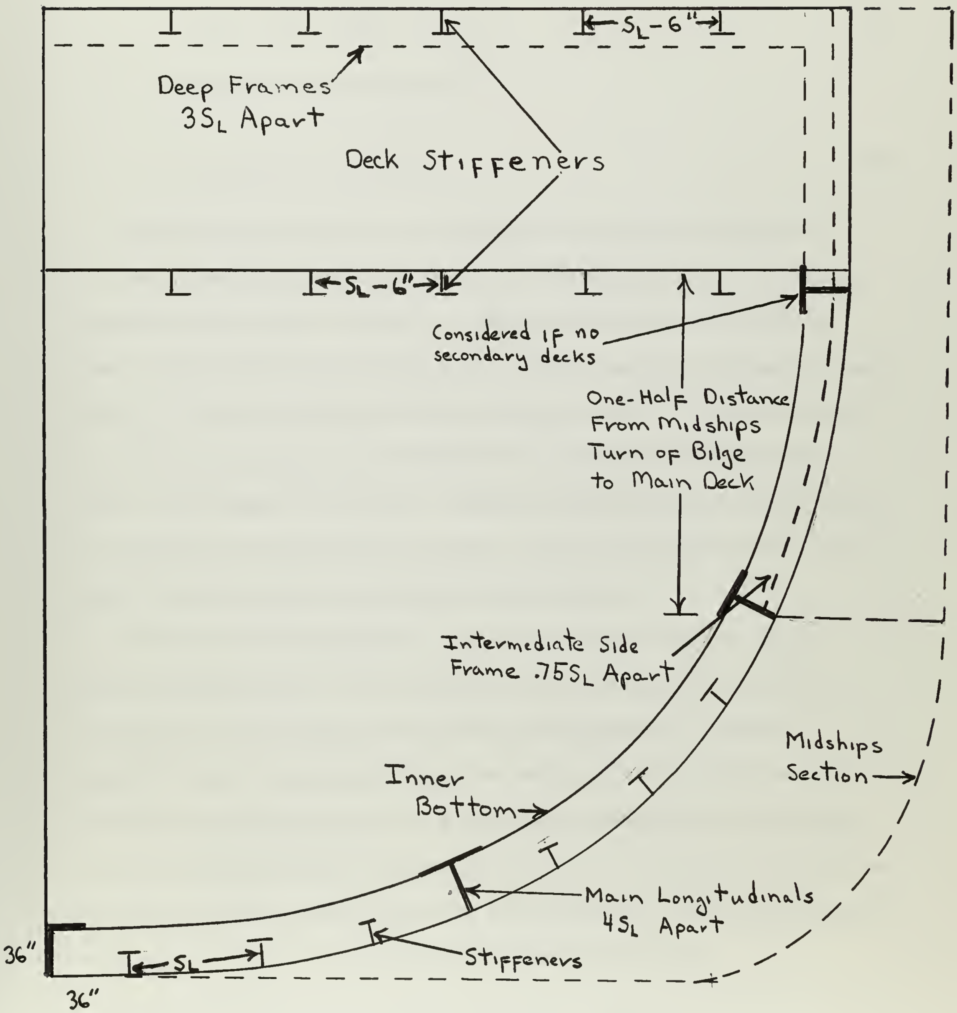
Structural Arrangement

One step in the design procedure will be to indicate a tentative outline of the section with which one is dealing. This will be unnecessary if it is only desired to check an existing design but in initial design some approach is needed. The designer may wish to simply revert to previous ship arrangements or may wish to invent his own arrangement. If, however, a quick, typical first arrangement guess is desired the use of Fig. (XI) in conjunction with the following short discussion is recommended.

Based on the Rules the following assumptions are made for initial frame spacing:

FIGURE XI

Tentative Structural Arrangement



1. Longitudinal frame spacing[28]

$$S_L = 20 + \frac{L}{40} \quad \text{(inches)} \tag{17}$$

2. Transverse frame spacing[29]

$$S_T = S_L + 3 \quad \text{(inches).} \tag{18}$$

Utilizing these equations the designer may now use Fig. (XI).

For a transversely framed ship it is not unreasonable to simply consider only the shell plating in longitudinal strength calculations when a first estimate is being made. In this case Eq. (18) may be used for S_T. Then the substitution of S_T for S_L in Fig. (11) and the use of the assumption regarding consideration of only shell plating would enable the designer to proceed. Smaller transverse web frames would be required and longitudinal stringers would be inserted purely on the basis of stability (plate buckling) considerations.

Note that this design is not completely divorced from midship section considerations. The midship section necessarily continues to dominate in the matter of structural arrangement. This fact, however, does not preclude either the design of some other section first and working backwards or a reduction or increase of scantlings over the midship section in some other section. It should be noted also that near the ships ends (away from the machinery spaces) other decks than the second may contribute to longitudinal strength.

28. American Bureau of Shipping, op. cit., Table A.

29. American Bureau of Shipping, op. cit., Table 12.

THE DESIGN METHOD

With the previous discussions in mind and in order to carry out the design of a section of a longitudinally or transversely framed ship, the steps which follow, and the equations and methods noted, are to be used in the order presented.

Previously Determined Data

1. Ship dimensions and coefficients, that is, L, B, D, H, Δ, and C_B should be assembled. It is assumed that the ship is of essentially welded construction.

2. Sketch an outline of the section of interest, indicating the positions of decks, pillars, hatches, transverse framing, bulkheads, and so forth. Use Fig. (XI) and associated discussion if desired.

3. Determine the material or materials which are to be utilized. Assume a corrosion allowance.

4. Set down any special requirements such as ice breaking bow, armor plating and so forth.

Procedure

1. Estimate the loads acting on the ship. This may be done by use of

$$M = \frac{\Delta L}{K} \tag{2}$$

and

$$V = \frac{\Delta}{K_v} \tag{3}$$

as previously discussed and by use of Fig. (I) which shows values of K derived from previous designs along with the values of K_v corresponding to these K values. These are entirely adequate for a first estimate.

2. Using

$$\sigma_1 = 1.19\sqrt[3]{L} \tag{7}$$

determine the acceptable safe bending stress. Note this σ_1 is for mild steel and if another material is used simply multiply this σ_1 by

$$\frac{\sigma_y \text{ (other material)}}{\sigma_y \text{ (mild steel)}} .$$

If σ_{ult} for the other material is only slightly greater than σ_y this relation may have to be modified. From these considerations and

$$\sigma_1 = \frac{M_y}{I} = \frac{M}{Z} \tag{5}$$

the section modulus may be determined. This will be a lower limit on Z.

3. Decide whether the hogging or sagging condition is to be investigated first. Both conditions should be checked but in general hogging is critical. Figure (I) plus the additional knowledge of where the machinery is located should aid in this decision.

4. At this point one of two paths may be followed.

a. It may be assumed that, in general,

$$.4 < \frac{Y_B}{D} < .5 \; .$$

Then using

$$Z = \frac{I}{Y_B} \; ,$$

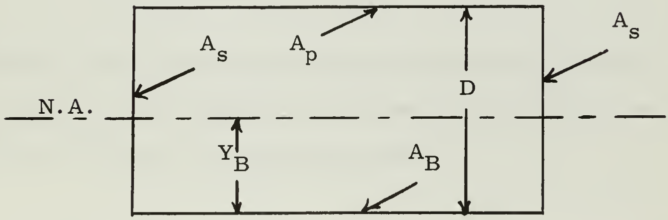
a required minimum value of I may be found. This may then be used to determine the value of σ_1 at any point in the section.

b. A simplified model of rectangular section may be used and the approximate formulas[30]

$$Y_B = \frac{D}{A} \left(A_s' + A_D\right) \tag{19}$$

$$I = D^2 \left[\frac{2A_s' \left(\frac{A}{3} - \frac{A_s'}{2} \right) + A_B \, A_D}{A} \right] \tag{20}$$

utilized. A_D, A_s' , and A_B values can be chosen to give required Z.



Method (a) is considered preferable by this author for other than the midships section.

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30. See appendix (A) for derivation and table of inertias format.

5. Using initially the longitudinal frame spacing determined from Fig. (XI) or from the designers arrangement the following may be determined directly or in terms of t for plating.[31]

a. $$\sigma_1 = \frac{M_y}{I} \qquad (5)$$

where M is at the section of interest and is based on a parabolic moment distribution. Use Fig. (II) to determine M based on M_{max} and section location along the length. Y is at the position of interest on the particular section and I is as determined in step (4) .

Primary stress will be taken as zero in the transverse[32] direction.

b. σ_2. This will be taken as 2000 psi in the longitudinal direction and 3000 psi in the orthogonal coplanar direction in any plate.

c. σ_3. Use

$$\sigma_3 = \frac{1}{288} K_L \; h \left(\frac{b_2}{t} \right)^2 \qquad (8)$$

with Fig. (IV) for both transverse and longitudinal directions. Take the head to twice the draft. Let the head be four feet for the main deck and two feet for intermediate decks.

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31. Longitudinals and plating are assumed fixed against lateral pressure and pinned in longitudinal bending.

32. The "transverse direction" is taken to mean that direction which is orthogonal to and coplanar with the longitudinal direction in the plane of the structural element considered.

d. τ. Find the shear stress by using

$$1. \quad \tau_{max} = 103.0 \frac{V}{tD} \text{ psi} \qquad (10)$$

in conjunction with

2. Figure (X)

in order to determine τ at any position in the section.

6. Add (superimpose) the stresses determined in step (5). Assume, as previously discussed, that all stresses have the same sign at some point in the member considered. Then let σ_x be the total longitudinal stress, σ_y the total transverse stress, and σ_{xy} the shear.

7. Using the stresses determined in step (6) and using the following expressions which are derived from Mohr's circle for σ_{max} and σ_{min} combined with the Mises-Hencky criteria[33] for limiting stress, determine the thickness of the plating under consideration.

See appendix (A) for derivation.

$$A^2 + 3B^2 = 1020 \times 10^6 \text{ psi}^{34} \qquad (21)$$

where

$$A = \frac{\sigma_x + \sigma_y}{2} \qquad (22)$$

and

$$B = \left[\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^{\;2} \right]^{1/2} \qquad (23)$$

8. Pause a moment and reflect on the factors of safety already used and to be used as this design develops.

---

33. See section on failure criteria.
34. See Appendix (A) for derivation of Eq. (21).

a. Equations (7), (10), and the Mises-Hencky criteria have been used to indicate safe values for stress values.

b. In stability considerations use will be made of

$$\frac{\tau_{cr}}{\tau} \geq 1.5$$

for shear where τ_{cr} is found from Eq. (13) and

$$\frac{\sigma_{cr}}{\sigma_w} \geq 1.5$$

for plate buckling and

$$\frac{\sigma_{cr}}{\sigma_w} \geq 1.75$$

for buckling of longitudinals where σ_w is the working stress and will be taken to be the longitudinal compressive stress which is acting (or tensile stress considered as compressive). Lateral pressure will be considered insignificant for purposes of instability considerations.

9. Using the transverse framing spacing determined from Fig. (XI) (note intermediate side frames run only through side plating) or the designers own arrangement instability criteria are applied.

Use

$$\tau_{cr} = \frac{K_v \pi^2 E}{12(1 - m^2)} \left(\frac{t}{b_2}\right)^2 \qquad (13)$$

with

$$\sigma_{cr} = \frac{K_b \pi^2 E}{12(1 - m^2)} \left(\frac{t}{b_1}\right)^2 \qquad (11)$$

or for transversely framed ships with

$$\sigma_{cr} = \frac{40,300}{1 + \frac{1}{950}\left(\frac{a_1}{t}\right)^{1.75}} \qquad (12)$$

in conjunction with

$$\left(\frac{\tau}{\tau_{cr}}\right)^2 + \frac{\sigma}{\sigma_{cr}} \leq 1 \qquad (14)$$

modified by step (8) to become

$$\left(1.5 \frac{\tau}{\tau_{cr}}\right)^2 + 1.5 \frac{\sigma}{\sigma_{cr}} \leq 1. \qquad (24)$$

Set Eq. (24) equal to one and solve for t. If it is less than the t determined in step (7) use step (7)'s t . Otherwise use this one.

10. Here again one of two paths may be followed depending on what choice was made in step (4).

a. It may be assumed that the plating constitutes approximately 80 percent of the section cross section area (this figure will obviously vary since 100 percent for transverse framing is sometimes reasonable) which is effective for longitudinal strength. If this assumption is made then once the thicknesses of all plating has been determined as in step (7), the required area of longitudinals may be

calculated. This area may then be divided up depending on type of framing desired above and below the neutral axis, frame spacing chosen, and the location of the neutral axis.

b. For any plating considered, for example the main deck, the area, Bt, may be calculated. The required sectional area of longitudinals is then $A_D - Bt$ and is divided among $(\frac{B}{b} - 1)$ longitudinals. The longitudinal dimensions may then be estimated.

11. The longitudinal stiffeners which have been tentatively determined should be checked now for stability. Suitable scantlings for the side intermediate frames may also be determined using lateral pressure considerations. For the longitudinals use Fig. (IX) with the area of longitudinal, a, determined in step (10) and I_2 determined from suitable section modulus graphs or from standard calculations [see appendix (A)] for beams. Use an initial effective width of plating of 60 t. When $(\sigma_{cr})_p$ and $(\sigma_{cr})_s$ have been determined and if $(\sigma_{cr})_s > (\sigma_{cr})_p$ the effective plate width, b_e, may be determined by setting

$$b_e = b\sqrt{\frac{(\sigma_{cr})_p}{(\sigma_{cr})_s}} \quad . \tag{25}$$

Using this value, I_L and a and hence a new value of $(\sigma_{cr})_s$ may be determined. This process should converge quickly to $(\sigma_{cr})_s \approx (\sigma_{cr})_p$. Determination of b_e other than 60 t is not, however, considered necessary in a first estimate.

Lateral, hydrostatic loading should also be checked but is again not considered critical in a first estimate except in the determination of intermediate side frame scantlings.

12. Since the critical buckling stress of the plating depends on the stiffeners remaining straight the following is used at this point. If for a given stiffener

$$\frac{EI}{b_1 D_s} \geq 21.5 \left(\frac{a_1}{b_1}\right)^2 - 7.5 \qquad (15)$$

where

$$D_s = \frac{E\,t^3}{12(1 - m^2)} \quad \text{(strut flexural rigidity)}$$

then the stiffener remains straight. If not,adjust stiffeners.

13. Steps (5) through (11) may have to be iterated in order to obtain scantlings of plating and longitudinals which meet both cross-sectional area and stability requirements and to optimize the design.

14. The steps to this point may be repeated (this time with fewer assumptions required since plating thicknesses and stiffener scantlings are available). This ends the initial design process which it is the purpose of this paper to present. It will be illustrated in a later example. Consider for a moment, however, the possible subsequent steps which will lead to a final solution.

15. The section should by now, be considerably modified from that initially assumed. Recalculate Z and ∇_1 more exactly. I and y_B can be calculated using the tabular form included in appendix (A).

16. Steps (5) through (14) can now be repeated for the more refined structure using exact formulae insofar as possible, allowing for all possible stress interaction combinations and checking and closely matching required safety margins.

17. These calculations should be carried out for a number of ship sections. Hopefully, this would lead to greater variation in scantlings than is current in ship structure. That is, scantlings reduced below those found amidships at least at sections within the middle one-half of the ship. This would aid considerably in optimizing hull weight while retaining the desired confidence in strength and durability.

18. Recalculate hull weight. Revise the shear force and bending moment distributions as a check on those previously assumed. Small modifications may be required.

It is noted that bulkheads, superstructure, dynamic effects, transverse strength, decks, and stem, and stern structure could have been considered here. However, in an initial design estimate this is not considered necessary from either the accuracy or speed of calculation viewpoint.

EXAMPLE OF DESIGN PROCEDURE

Discussion

Now that the basic formulae and criteria are laid down and the design procedure set forth and following the example of St. Denis[35], it would seem the most lucid at this point to work out a numerical example. If it is then desired to develop or check the design of any particular ship this may readily be done simply by following through this example with one's own numbers. Since the purpose of this paper is to extend the midship section method to permit accommodation of shearing forces acting in addition to σ_1, σ_2, and σ_3 some structural parts of a ship whose loading is not subject to the additional effects of shearing may be considered in this example for completeness in determination of longitudinal strength but not in any other part of this paper. Superstructure will not be considered in this example. The example will be carried through for longitudinal framing. By taking note of the previous brief discussion of framing systems the transition to transverse framing may be readily made. Thicknesses are left as found. . The designer will need to choose standard commercial sizes.

Previously Determined Data

1. Ship Dimensions

L = 375 ft.	D = 23 ft.
H = 15 ft.	B = 40 ft.
Δ = 3500 tons	

H, D, and B are at the section of interest.

35. St. Denis, *op. cit.*

Spacing of transverse machinery bulkheads is 40 ft. considering L/4 section.

2. Using Fig. (XI) and the discussion preceding it, from Eq. (17)

$$S_L = 20 + \frac{375}{40} = 29.35 \text{ inches} \approx 30 \text{ inches.}$$

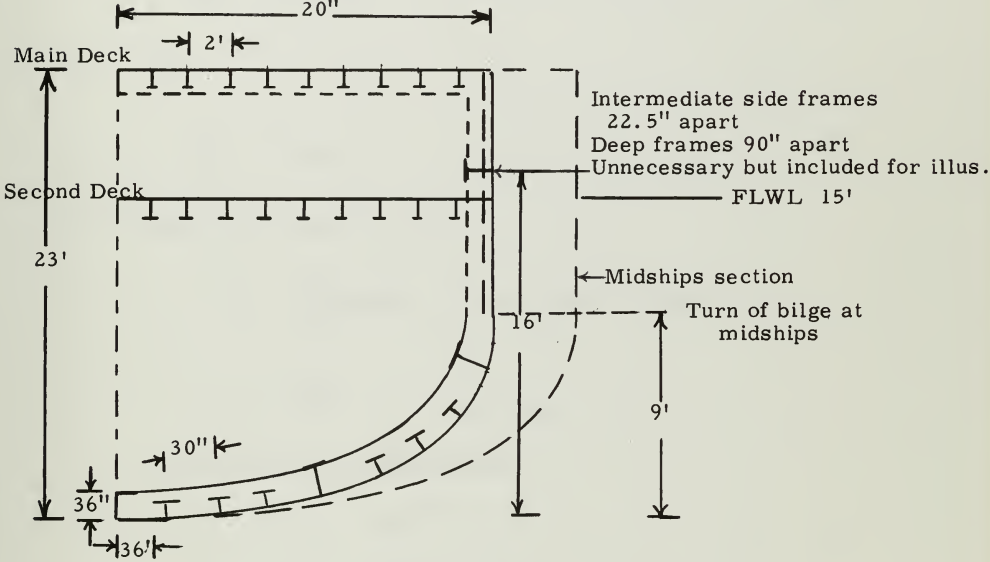


FIGURE XII

3. The material to be used in this example will be medium steel throughout with

σ_y = 35,000 psi (tension and compression)

m = .3

τ_y = 20,000 psi.

Corrosion allowance: .125 in. external
.075 in. internal

4. No special requirements exist.

Procedure

1. Loads acting on ship are,

using

$$\frac{\Delta L}{K} = M_{max} \tag{2}$$

and

$$\frac{\Delta}{K_v} = V_{max} \tag{3}$$

with Fig. (I) and Fig. (II)

$$M = \frac{3500 \times 375}{27} \times .55 = 26,800 \quad \text{ft - tons}$$

$$V = \frac{3500}{8} \times 1.0 = 437 \text{ tons.}$$

2. Maximum safe bending stress

$$\sigma_1' = 1.19 \sqrt[3]{L} \tag{7}$$

$$\sigma_1' = 1.19 \sqrt[3]{375}\ (2240) = 19,150 \text{ psi}$$

Then if

$$\sigma_1 = \frac{M}{Z}$$

the limiting Z based on σ_1' is

$$Z = \frac{26,800}{19,150}\ (2240) = 3120 \text{ in.}^2 \text{ - ft.}$$

3. The hogging condition is to be investigated.

4a. Determination of minimum required I.

Let $\frac{Y_B}{D} = .45$

then $Y_B = .45(23) = 10.4$ ft.

If $Z = \frac{I}{Y_B}$

then $I = 3120 \times 10.4 = 32,600$ in.^{2} - ft.^{2}.

From this point individual ship sections may be considered.

<u>Side Shell at Neutral Axis</u>

5. a. $\sigma_1 = \frac{M_Y}{I}$

$= \frac{M(0)}{I} = 0$

b. $\sigma_2 = 3000$ psi (transverse)

$\sigma_2 = 2000$ psi (longitudinal)

c. Using Eq. (8) and Fig. (IV) and

$a_2 = 96''$

$b_2 = 22.5''$

$\frac{a_2}{b_2} = \frac{96}{22.5} = 4.25$

$h = 2H - 2Y_B$

$= 30 - 10.4 = 19.6$ ft.

it is seen that

$$K_{L_1} = 1$$

$$K_{L_2} = .685$$

and thus

$$\sigma_3 \text{ (longitudinal)} = \frac{1.0}{288} \times 64 \times 19.6 \left(\frac{22.5}{t} \right)^2$$

$$= \frac{2130}{t^2} \text{ psi}$$

$$\sigma_3 \text{ (transverse)} = \frac{.685}{288} \times 64 \times 19.6 \times \left(\frac{22.5}{t} \right)^2$$

$$= \frac{1510}{t^2} \text{ psi} .$$

d. Using Eq. (1) and Fig. (V)

$$\tau_{max} = 103.0 \times \frac{437}{t \times 23}$$

$$= \frac{1950}{t} \text{ psi}$$

τ = 100 percent of τ_{max} therefore

$$\tau = \frac{1950}{t} \text{ psi} .$$

6. Superimposing stresses it is seen that stresses are

	Transverse	Longitudinal
σ_1	0	0
σ_2	3000	2000
σ_3	$\frac{1510}{t^2}$	$\frac{2130}{t^2}$

$$\sigma_y = 3000 + \frac{1510}{t^2} \qquad \sigma_x = 2000 + \frac{2130}{t^2}$$

$$\tau_{xy} \quad \frac{1950}{t} \text{ psi}$$

7. Using Eq. (21) and solving for t

$$\left(\frac{5000}{2} + \frac{3640}{2t^2}\right)^2 + 3\left(-\frac{1000}{2} + \frac{620}{2t^2}\right)^2 + 3\left(\frac{1950}{t}\right)^2 = (35 \times 10^3)^2$$

t = .260 + .125 (corrosion allowance)

t = .385".

<u>Side Plating at Shear Strake</u>

5. a. $\sigma_1 = \frac{M_Y}{I}$

then with Y = 12.6 ft. (upper deck edge)

$$\sigma_1 = \frac{26,800 \times 12.6}{32,600} \times 2240 = 23,200 \text{ psi (tension).}$$

b. σ_2 = 3000 psi (transverse)

σ_2 = 2000 psi (longitudinal).

c. a_2 = 96" (8 ft.) $\quad \frac{a_2}{b_2} = \frac{96}{22.5} = 4.25$

b_2 = 22.5" $\quad h = 30 - 23 = 7$

then $K_{L_1} = 1$

$K_{L_2} = .685$

and thus

$$\sigma_3 \text{ (longitudinal)} = \frac{1.0}{288} \times 64 \times 7 \times \left(\frac{22.5}{t}\right)^2$$

$$= 760/t^2 \text{ psi}$$

$$\sigma_3 \text{ (transverse)} = .685 \times 64 \times 7 \times \left(\frac{22.5}{t}\right)^2$$

$$= 540/t^2 \text{ psi}.$$

d. Using Eq. (1) and Fig. (V)

$$\tau_{max} = \frac{1950}{t}$$

τ = 80 percent of τ_{max} therefore

$$\tau = .8 \times \frac{1950}{t} = \frac{1560}{t} \text{ psi}.$$

6. Superimposing stresses it is seen that stresses are:

	Transverse	Longitudinal
σ_1	0	23,200
σ_2	3000	2000
σ_3	$540/t^2$	$760/t^2$

$\sigma_y = 3000 + 540/t^2$ $\qquad$ $\sigma_x = 25,200 + 760/t^2$

τ_{xy} $\qquad \frac{1560}{t}$

7. Using Eq. (21) and solving for t

$$\left(\frac{28,200}{2} + \frac{1300}{2t^2}\right)^2 + 3\left[\left(\frac{220}{2t^2} - \frac{22,200}{2}\right)^2 + \left(\frac{1560}{t}\right)^2\right] = 1225 \times 10^6$$

$t = .35 + .125$ (corrosion allowance).

$= .475$ inches .

Main Deck

5. a. $\sigma_1 = 23,200$ psi (same as sheer strake in this example)

b. $\sigma_2 = 3000$ psi (transverse)

$\sigma_2 = 2000$ psi (longitudinal)

c. $a_2 = 90''$

$b_2 = 24''$ $\qquad$ $h = 4$ ft.

$\frac{a_2}{b_2} = \frac{90}{24} = 3.75$

$$K_{L_1} = 1.0$$

$$K_{L_2} = .685$$

$$\sigma_3 \text{ (longitudinal)} = \frac{.685}{288} \times 64 \times 4 \left(\frac{24}{t}\right)^2$$

$$= \frac{350}{t^2} \text{ psi}$$

$$\sigma_3 \text{ (transverse)} = \frac{1.0}{288} \times 64 \times 4 \left(\frac{24}{t}\right)^2$$

$$= \frac{510}{t^2} \text{ psi .}$$

d. $\tau = \frac{1560}{t}$ psi (same maximum as sheer strake).

6. Superimposing stresses it is seen that stresses are:

	Transverse	Longitudinal
σ_1	0	23,200
σ_2	3000	2000
σ_3	$\frac{510}{t^2}$	$\frac{350}{t^2}$
	$\sigma_y = 3000 + \frac{510}{t^2}$	$\sigma_x = 25,200 + \frac{350}{t^2}$
τ_{xy}	$\frac{1560}{t}$	

7. Using Eq. (21) and solving for t

$$\left(\frac{28,200}{2} + \frac{860}{2t^2}\right) + 3\left[\left(-\frac{160}{2t^2} + \frac{22,200}{2}\right)^2 + \left(\frac{1560}{t}\right)^2\right] = 1225 \times 10^6$$

$t = .342 + .125 = .457$ inches.

Bottom Plating

For this example consider the stresses at a point on the plating three feet above the baseline.

5. a. $\sigma_1 = \frac{M_Y}{I}$

$$= \frac{26,800 \times 7.4}{32,600} \times 2240 = 13,600 \text{ psi (compression)}$$

b. $\sigma_2 = 3000$ psi (transverse)

$\sigma_2 = 200$ psi (longitudinal)

c. $a_2 = 90$ $\qquad K_{L_1} = 1.0$

$b_2 = 30$ $\qquad K_{L_2} = .685$

$\frac{a_2}{b_2} = \frac{90}{30} = 3$ $\qquad h = 30 - 3 = 27$

$$\sigma_3 \text{ (longitudinal)} = \frac{.685}{288} \times 64 \times 27 \left(\frac{30}{t}\right)^2$$

$$= 3640/t^2 \text{ psi}$$

$$\sigma_3 \text{ (transverse)} = \frac{1.0}{288} \times 64 \times 27 \left(\frac{30}{t}\right)^2$$

$$= 5300/t^2 \text{ psi}$$

d. Using Eq. (1) and Fig. (V)

$$\tau_{max} = \frac{1950}{t}$$

$\tau \approx$ 80 percent of τ_{max} therefore

$$\tau = .8 \times \frac{1950}{t} = \frac{1560}{t} \text{ psi}$$

6. Superimposing stresses it is seen that stresses are:

	Transverse	Longitudinal
σ_1	0	13,600
σ_2	3000	2000
σ_3	$5300/t^2$	$3640/t^2$
σ_y	$= 3000 + 5300/t^2$	$\sigma_x = 15,600 + 3640/t^2$
τ_{xy}	$\frac{1560}{t}$	

7. Using Eq. (21) and solving for t

$$\left(9,300 + \frac{4470}{t^2}\right)^2 + 3\left[\left(6300 - \frac{830}{t^2}\right)^2 + \left(\frac{1560}{t}\right)^2\right] = 1225 \times 10^6$$

t = .42 + .125 (corrosion allowance)

= .545 inches .

Second Deck

5. a. $\sigma_1 = \dfrac{26,800 \times 4.6}{32,600} \times 2240 = 8450$ psi

b. $\sigma_2 = 3000$ psi transverse; $\sigma_2 = 2000$ psi longitudinal

c. let a_2/b_2 be greater than 3.0 then

$K_{L_1} = 1.0$

$K_{L_2} = .685$

$$\left.\begin{array}{l} \sigma_3 \text{ (longitudinal)} = \dfrac{1050}{t^2} \text{ psi} \\ \\ \sigma_3 \text{ (transverse)} = \dfrac{1530}{t^2} \text{ psi} \end{array}\right\} \text{(head to main deck) + 4 ft. = h}$$

d. $\tau_{max} = \dfrac{1950}{t}$, from Fig. (X) $\tau = \dfrac{.85 \times 1950}{t} \approx \dfrac{1700}{t}$ psi

6. Superimposing stresses it is seen that stresses are:

	Transverse	Longitudinal
σ_1	0	8450
σ_2	3000	2000
σ_3	$\dfrac{1530}{t^2}$	$\dfrac{1050}{t^2}$
	$\sigma_y = 3000 + \dfrac{1530}{t^2}$	$\sigma_x = 10,450 + \dfrac{1050}{t^2}$
τ_{xy}	$\dfrac{1700}{t}$	

7. Using Eq. (21) and solving for t

$$\left(\frac{13,450}{2} + \frac{1530}{2t^2}\right)^2 + 3\left[\left(\frac{7450}{2} - \frac{1050}{2t^2}\right)^2 + \left(\frac{1700}{t}\right)^2\right] = 1225 \times 10^6$$

t = .17 + .075 (corrosion allowance) = .245.

Inner Bottom

5. a. σ_1 = 13,600 psi (as calculated for bottom plating)

b. σ_2 = 3000 psi transverse

σ_2 = 2000 psi longitudinal

c. $\frac{a_2}{b_2} = \frac{90}{30} = 3$

$K_{L_1} = 1.0$

take head to main deck
h = 20 ft.

$K_{L_2} = .685$

$$\sigma_3 \text{ (longitudinal)} = \frac{.685}{288} \times 64 \times 20 \left(\frac{30}{t}\right)^2$$

$$= \frac{2700}{t^2} \text{ psi}$$

$$\sigma_3 \text{ (transverse)} = \frac{3920}{t^2} \text{ psi}$$

d. $\tau = .65 \times \frac{1950}{t} = \frac{1270}{t}$ psi

6. Superimposing stresses it is seen that stresses are:

	Transverse	Longitudinal
σ_1	0	13,600
σ_2	3000	2000
σ_3	$3920/t^2$	$2700/t^2$
σ_y	$= 3000 + 3920/t^2$	$\sigma_x = 15,600 + 2700/t^2$
τ_{xy}	$1270/t$	

7. Using Eq. (21) and solving for t

$$\left(\frac{18,600}{2} + \frac{5620}{2t^2}\right)^2 + 3 \quad \left(\frac{12,600}{2} - \frac{1220}{2t^2}\right)^2 + \left(\frac{1270}{t}\right)^2$$

$t = .344 + .125 = .469$

8. Applying instability criteria to <u>Side Shell at Neutral Axis</u>, it is seen that

$$\tau = \frac{1950}{t} \text{ psi from previous calculations.}$$

τ_{cr} for simply supported edges using Eq. (13) with

$$K_v = 5.35 + \left(\frac{22.5}{84}\right)^2 = 5.42$$

and

$$b_2 = 22.5 \text{ inches}$$

is

$$\tau_{cr} = 27.1 \times 10^6 \times 5.4 \left(\frac{t}{22.5}\right)^2$$

$$= 28.9 \times 10^4 \, t^2 \text{ psi}$$

$$\sigma_1 = 0$$

Then plugging

$$\left(1.5 \frac{\tau}{\tau_{cr}}\right)^2 + 1.5 \frac{\sigma}{\sigma_{cr}} \leq 1 \qquad (14)$$

With τ_{cr} and σ_1 it is seen that if

$$\left(1.5 \times \frac{1950}{t} \times \frac{1}{28.9 \times 10^4 \, t^2}\right)^2 = 1$$

$$t = .212 + .125 \text{ inches}$$

$$= .337 \text{ inches.}$$

This is less than .385 inches therefore chose

$$t = .385 \text{ inches.}$$

<u>Side Plating at Sheer Strake</u>

$\tau = \frac{1560}{t}$ psi from previous calculations. τ_{cr} for simply supported edges using Eq. (13) with

$$K_v = 5.4 \text{ and } b_2 = 22.5 \text{ inches}$$

is

$$\tau_{cr} = 28.9 \times 10^4 \, t^2 \text{ psi}$$

$$\sigma_1 = 23,200 \text{ psi (taken as compression)}$$

$$\frac{a_1}{b_1} = \frac{22.5}{84} \quad \text{therefore using Eq. (12)}$$

$$\sigma_{cr} = \frac{40,300}{1 + \frac{1}{950}\left(\frac{22.5}{t}\right)^{1.75}} = 20,800 \text{ psi}$$

Let $t = .475$, then using Eq. (14)

$$\left(\frac{1.5 \times 3060}{62,400}\right)^2 + \frac{3.48 \times 10^4}{2.08 \times 10^4} > 1.$$

Thus buckling occurs with $t = .475$. Two choices are open here. Thickness can be increased or framespacing changed. In this case change thickness to .7 inches. Now the

Factor of Safety = 1.2.

This is less than the 1.5 criteria previously set up. Perhaps some dishing of plating would be permissible. At this stage of design take

$t = .7$ inches.

<u>Main Deck</u>

$\tau = \frac{1560}{t}$ from previous calculations. τ_{cr} for simply supported edges using Eq. (13) with

$$K_v = 5.35 + 4\left(\frac{1}{4}\right)^2 = 5.55$$

$$\tau_{cr} = 27.1 \times 10^6 \, t^2 \text{ psi}$$

$$\sigma_1 = 23,200 \text{ psi (taken as compression)}$$

$$\frac{a_1}{b_1} = \frac{90}{24} \quad \text{therefore} \quad K_b = 4.0$$

Using Eq. (11)

$$\tau_{cr} = 18.7 \times 10^4 \, t^2$$

Then use Eq. (14)

$$\left(\frac{1.5 \times 1420 \times 10^{-4}}{t^3 \, 27.1}\right)^2 + \left(\frac{2.32}{18.7 \, t^2}\right) \leq 1.$$

For $t = .467$ this is ≤ 1 therefore choose

$$t = .467 \text{ inches.}$$

Bottom

$\tau = \frac{1560}{t}$ psi from previous calculations. τ_{cr} for simply supported edges using Eq. (13) with

$$K_v = 5.35 + 4 \left(\frac{1}{3}\right)^2 = 5.79 \; ; \quad b_2 = 30''$$

is

$$\tau_{cr} = 17.4 \times 10^4 \, t^2 \text{ psi.}$$

$$\tau_1 = 13{,}600 \text{ psi.}$$

$$\frac{a_1}{b_1} = \frac{90}{30} = 3 \quad \text{therefore} \quad K_b = 4.0.$$

Using Eq. (11)

$$\tau_{cr} = 12.0 \times 10^4 \, t^2 \text{ psi.}$$

Then use Eq. (14)

$$\left(\frac{1.5}{17.4} \times \frac{1560}{t^3} \times 10^{-4}\right)^2 + \left(1.5 \times \frac{13,600}{12.0 \times 10^4\ t^2}\right) \leq 1.$$

If $t = .545$ inches is used this is ≤ 1. Thus choose

$t = .545$ inches.

Inner Bottom

Calculations are identical to bottom structure. Eq. (14) ≤ 1. Choose

$t = .469$ inches.

Second Deck

Calculations identical to main deck with transverse stiffeners at 90 inch intervals. For this example Eq. (14) is ≤ 1. The ideal situation is for the chosen scantlings to exactly satisfy both the yield and instability criteria. Choose

$t = .255$ inches.

9. a. Find required area of longitudinals based on assumption that the plating consititutes 80 percent of area.

Plating Area

Side Shell	2 x 8' x 12" x .385"	=	73.6 in.^{2}
Sheer Strake	2 x 6' x 12" x .70"	=	97.5
Main Deck	2 x 20' x 12" x .467"	=	222.0
Second Deck	2 x 20' x 12" x .255"	=	122.5
Bottom	2 x 21' x 12" x .545"	=	275.0
Inner Bottom	2 x 21' x 12" x .469"	=	235.0
			1025.6 in.^{2}

Then total area required is

$$\frac{1025.6}{.8} = A$$

$$A_T = 1280.0 \text{ in.}^2 .$$

Thus for the longitudinals an area of

$$A_L = 1280 - 978.7 = 301.3 \text{ in.}^2$$

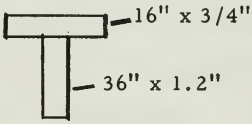
is required.

In this example this is divided between

7 main longitudinals and

52 longitudinal stiffeners.

As a first estimate take main longitudinals as



main longitudinal

$$A_{Tm} = 7 \times 16 \times .75 + 7 \times 36 \times .5 = 210.0 \text{ in.}^2.$$

Then for the stiffeners it is seen that

$$A_{Ts} = 301.3 - 210.0 = 91.3 \text{ in.}^2.$$

<u>This seems small</u>. Reversing this process take the side stringers and deck and bottom longitudinals to be 1/2 - 10" x 2-3/4" x 9# Jr. B'm (4.5#) [1.32""]. Then

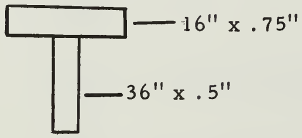
$$A_{Ts} = 54 \times 1.32 = 71.3 \text{ in.}^2.$$

Then for main longitudinals

$$A_{Tm} = 301.3 - 71.3 = 230 \text{ in.}^2$$

$$\frac{A_m}{7} = 32.8 \text{ in.}^2.$$

Choose main longitudinals as shown $A_m = 30$ sq. in.



Longitudinal stiffeners 1/2 - 10" x 2.75" x 9# Jr. B'm (4.5 #)

[1.32" "].

Area required is met by these choices within 7 x 2.8" = 19.6 in.^{2}.

Since the deep web main longitudinal so near the second deck is not necessary in this case then

$$30 \text{ in.}^2 - 19.6 \text{ in.}^2 = 10.4 \text{ in.}^2.$$

or the required area is exceeded by 10.4 in.^{2}.

10. Determine the buckling strength of all longitudinal stiffeners using Fig. (IX) and data from step (9).

Longitudinal stiffeners	a	I
main deck	11.58	20.0
second deck	5.12	16.5
bottom	18.52	21.0
side	9.47	19.0

Main Deck

Assume simple support K = 1

$$C = \frac{90}{\sqrt{\frac{20}{11.58}}} \sqrt{\frac{15.5}{13,500}} = 2.24$$

$$F = \frac{\sigma_{cR}}{\sigma_{cr}} = .83$$

$$\sigma_{cR} = .83 \times 15.5 \times 2240 = 28,000 \text{ psi}$$

$$\sigma_1 = 23,200 \text{ psi}$$

Factor of safety = 1.2 against buckling. This is too low. Increase stiffener to 1/2 - 12" x 3" x 11.8# Jr. B'm then a = 12.00 in.^{2}

I = 36 C = 1.7 F = .95 σ_{cR} = 32,200.

Factor of safety = 1.44 which is close enough to 1.5 factor for plating for a first estimate, although below the desired 1.75.

Second Deck

C = 1.65

F = .96 σ_1 = 8450 psi

$$\sigma_{cR} = .96 \times 15.5 \times 2240 = 33,200 \text{ psi.}$$

$$\text{Factor of safety} = \frac{33.2}{8.4} = 3.96$$

This is high. Reduce scantlings of longitudinals to 1/2 - 7" x 2 1/8" x5.5# Jr. B'm [.805 " "].

$a = 11.06 \text{ in.}^2$ $\qquad$ $I = 5$

$C = 4.4$ $\qquad$ $F = .41$

Factor of safety = 1.7 which is close enough to 1.75 for this first estimate pending more exact later calculations.

Bottom

$C = 2.78$

$F = .72$

$\sigma_{cR} = 25,000$

F. S. = 1.84 which is close enough to 1.75 until later calculation involving both lateral and axial loading.

Side

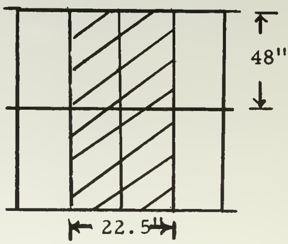
$C = .52$

$F > 1.0$ $\qquad$ $\sigma_{cR} = 35,000$

F. S. (based on max σ_1) $>$ 1.75. Longitudinals are adequate. The side longitudinals, however, serve here to reduce the b_1 dimension of the plate.

Check Intermediate Side Frame

Assume that a pressure of 7 psi (reasonable as a quick calculation will show) acts uniformly over plating



Assuming Ends Fixed

$$M_{max} = \frac{WL^2}{12} = \frac{7 \times 22.5 \times (48)^2}{12 \times 2240}$$

$$\sigma_{max} = \frac{M}{Z} = \frac{7 \times 22.5 \times (48)^2}{12 \times 2240 \times Z} = \frac{13.4}{Z} .$$

If $\quad F.S. = 1.5 = \dfrac{15.5}{\dfrac{13.4}{Z}}$

then $\quad Z = 1.5 \times \dfrac{13.4}{15.5} = 1.3$ which is very small.

What if $\quad L = 96$? Then $Z = 5.2$ which is reasonable.

Choose

1/2 - 8" x 4" x 13# light B'm (6.5#) (1.91 in.^{2}) ($Z = 5.4$)

All of plating is effective. Factor of safety $>$ 1.75.

11. The stiffeners remain straight.

12. With scantlings now available check whether sufficient area remains available to meet section modulus requirements. If not, adjust.

13. As may be seen adjustments to match factors of safety and insure meeting all criteria have been necessary. The main section scantlings are now available. Deep frame scantlings must be obtained from a transverse strength calculation. Reasonably accurate calculations for I, τ_{xy}, Y_B, and so forth may now be made without recourse to plots for approximations.

CONCLUSIONS AND RECOMMENDATIONS

The sections of a paper usually named "Conclusions" and "Results" and "Discussion of Results", are not needed in this paper. All of these headings are really treated in the design method which was developed and which is thus both a conclusion, a result, and the example a discussion of the results. This author, however, wishes to reiterate here that this report is intended for an approximate initial estimate of other than midship section scantlings. It can also be used to check the structural reliability of a section. In its simplicity it is felt that it should encourage at least a check on sections other than midships in preliminary design studies.

RECOMMENDATIONS

There are many recommendations which might be made in the entire field of structural design of ships. This discussion, however, will be limited to improvements which might be made on the design method considered here.

It is recommended that as the methods of plastic design as opposed to elastic are developed that these be applied to arbitrary section analysis in order to give more realistic factors of safety. This could lead to lighter weight design and will certainly yield more nearly true information.

Instrumentation of actual ships in order to obtain more exact information concerning the distribution of shear around the ship's peripherey is recommended. In addition, more data on bending moment and shear distributions in actual ships would be useful, particularly as relates to the maximum conditions.

It is recommended that a simplified method or means of approximation for determining the contribution of superstructure to longitudinal strength be considered.

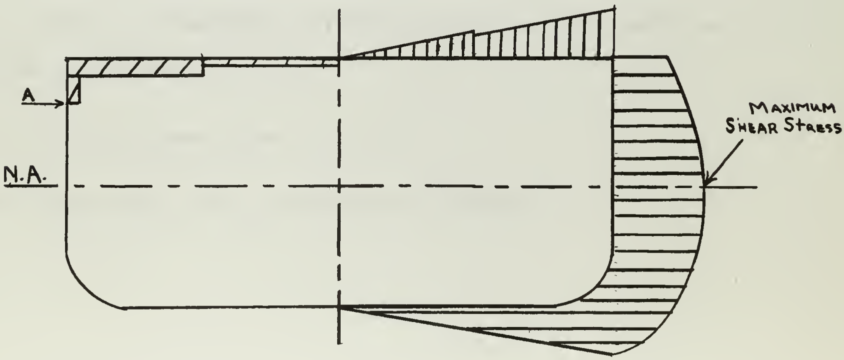
APPENDIX A

APPENDIX A
SUPPLEMENTARY DISCUSSION

1. General Analysis of Plating Stresses

a. Discussion - Initially it was hoped that this paper might set forth in terms of some one dimension, say ship length, L, a set of very general equations which would enable the designer to very quickly arrive at an at least approximate initial solution for a ship's section. This approach, although attacked from several angles, proved to be so approximate in all instances as to be essentially worthless. There were some pieces of these attempts, in particular for the L/4 section of a ship, which if not particularly useful, at least illustrate some of the very broad assumptions attempted by this author for "the range of ships," and the initial thoughts which he had on the subject at hand. For this reason these pieces are included as an appendix and while they may prove to be of no particular use to the designer it is not felt that they are completely lacking in interest.

b. Side plating - This would seem to be a logical starting point for an analysis since if one looks at the distribution of shear as a simple ship section, viz,



it is seen that the maximum shear occurs at the neutral axis in the side plating and is relatively large at both the upper deck edge and turn of bilge. As to which of these portions should be checked to see if maximum stress conditions exist, this author sees no choice. All three positions must be checked. This reasoning is based on the fact that the shear stress at the turn of bilge will, since in general the neutral axis is closer to the bilge or bottom than the main deck, be larger than that at the main deck. The same reasoning also indicates that the primary stress will be smaller at the turn of bilge that at the upper deck edge. The tertiary stress will be larger at turn of bilge than at any point above. Thus it is reasonable to assume that no honest assumption can be made concerning combinations of stresses in the side plating. Check all three positions.

At the neutral axis there are three stresses acting in combination, shear, tertiary, and secondary. Primary stress is zero. Make the following assumptions

1. The maximum shear force, V, is $\Delta/9$.

2. Secondary stress, σ_2, is constant at 2000 psi.

3. Tertiary stress, σ_3, is given by

$$\sigma_3 = \frac{1}{288} K_L \rho \, h \left(\frac{b_2}{t}\right)^2 \qquad (8)$$

where K_L is obtained from Fig. (IV).

With these assumptions it is seen that if

$$V_{max} = \frac{\Delta}{9}$$

then

$$\tau_{max} = \frac{2240 \, VQ_{max}}{12 \, t \, I} = \frac{2240}{12} \times \frac{\Delta}{9} \times \frac{Q_{max}}{t \, I}$$

$$= 20.8 \, \frac{LBHC_B}{t \, I} \, Q_{max} \qquad (26)$$

Breaking Eq. (26) down even further, viz,

$$\tau_{max} = \frac{2240 \, V}{12 \, A_w} \times \text{constant}$$

and if a value of 1.1 is chosen for the constant as was done in the body of this paper then

$$\tau_{max} = .298 \, \frac{LBHC_B}{t \, D} \, . \qquad (27)$$

It is noted that Eq. (27) is less general than Eq. (26) in that the simplification of Q and I based on a beam eliminates consideration of longitudinals. Also, replacing Q_{max} by Q in Eq. (26) defines τ.

It is seen then that equations exist for the stresses but without taking some typical examples and converting Eq. (8) and (27) into numerical values and using them in conjunction with the constant secondary stress there is still have very little to go on. It may be possible to reduce all three stresses to a common form.

Assume that Eq. (27) is suitable for illustrative purposes. It is further assumed that

1. $H = .65D$ as an average over the range of ships, fully loaded.

2. $B = 2H$ as an average over the range of ships, fully loaded.

3. $C_B = .75$. This is fairly high for the range of ships and should therefore be conservative. With these assumptions Eq. (27) becomes

$$\tau_{max} = .290 \frac{LH}{t} \quad . \tag{28}$$

In the case of the tertiary stress, Eq. (8), it is assumed that

1. $K_{L_1} = 1.0$ and $K_{L_2} = .685$. That is to say that in Fig. (IV) a_2/b_2 will be greater than two (2).

2. $\rho = 64 \frac{\text{lbs.}}{\text{ft.}^3}$

3. The neutral axis is, in general, forty percent of the depth above the baseline. Then

$$h = .65D - .4D = .25D$$

and since it has been assumed that $D = \frac{H}{.65}$ then

$$h = \frac{.25}{.65} H = .384 H \approx .4H.$$

This does not allow for wave crests as is done by St. Denis[36].

Introducing these assumptions into Eq. (8) it is seen that for K_{L_1} or K_{L_2}

$$\sigma_3 \approx (.09 \text{ or } .06)H \left(\frac{b_2}{t}\right)^2 . \tag{29}$$

Thus, in summary, at the neutral axis the following equations apply

$$\tau = .290 \frac{LH}{t} \text{ psi} \tag{28}$$

$$\sigma_1 = 0$$

$$\sigma_2 = 2000 \text{ psi}$$

$$\sigma_3 = (.09 \text{ or } .06)H \left(\frac{b_2}{t}\right)^2 . \tag{29}$$

Before any discussion of t, b_2, and H is attempted the additional assumptions used to develop equations of the type above will be listed and the equations summarized. It is understood that the assumptions are applied in the same manner as before.

At the turn of bilge

1. The shear stress Eq. (28) is reduced by 25 percent due to the

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36. St. Denis, op. cit., p. 51.

distribution of shear over the cross section. Examination of most ship sections will show this to be conservative.

2. h = .6H.

3. Primary stress must be considered here.

At the L/4 section assume

$$M = .55 \left(\frac{\Delta L}{35}\right)$$

$$\cong \frac{\Delta L}{64}$$

Then if

$$\sigma_1 = \frac{M_y}{I} \qquad (5)$$

is combined with

$$\sigma_1 = 1.19 \sqrt[3]{L} \qquad (7)$$

it is seen that

$$\sigma_1 \cong .33 \sqrt[3]{L} \qquad (30)$$

at the turn of bilge assumed at .2H above the baseline. With these assumptions and the equation given the stresses then at the turn of bilge may be shown to be

$$\sigma_1 = .33 \sqrt[3]{L} \quad \text{tsi} \qquad (30)$$

$$\sigma_2 = 2000 \text{ psi}$$

[illegible]

$$[illegible]$$

[illegible]

$$[illegible]$$

[illegible]

$$[illegible]$$

$$\sigma_3 = (.135 \text{ or } .094)\left(\frac{b_2}{t}\right)^2 \text{ psi} \tag{31}$$

$$\tau = .210 \frac{LH}{t} \text{ psi} \tag{32}$$

The same assumptions but with modified σ_1 and τ stresses due to location yield at the sheer strake, upper deck edge

$$\sigma_1 = .66 \quad {}^{3}\; L \tag{33}$$

$$\sigma_2 = 2000 \text{ psi}$$

$$\sigma_3 = 0$$

$$\tau = .19 \frac{LH}{t} \text{ psi} \tag{34}$$

The stresses existing in the main deck (that is the maximum stresses) will be assumed the same as those in the shear strake.

At this point the reader is no doubt a little uneasy. The equations developed to indicate shear stress magnitudes are obviously very approximate. Before these can be used in a Mohr's circle analysis for maximum and minimum principle stresses, however, and based on only one known ship's dimension, more assumptions need to be made. The following are typical of the attempts made to eliminate all but the dimension L:

1. Extrapolating from Evans [37]

37. Evans, <u>op. cit.</u>, p. 253.

$$L = 22 (H - 4.5)$$

and this is arbitrarily reduced to

$$L = 19H \tag{34}$$

which when the assumption

$$\frac{L}{H} = 16.7$$

of table (A) of the Rules and

$$\frac{L}{H} = 20$$

of table (12) of the Rules are considered is not unreasonable.

2. Let $b_2 = 60t$.

3. Assume some typical values of t.

For example the Rules yield for the sheer strake

$$t_t = .27 + \frac{L}{100} \times .09 \text{ inches}$$

$$t_L = .20 + \frac{L}{100} \times .10 \text{ inches.}$$

4. The material is mild steel. This limitation (it limits σ_1 only in this development, however) is too demanding.

It has become obvious at this point that the author has almost assumed his way out of reality. In only one respect did this discussion yield anything other than a few ideas on some approximations which might be used to obtain rough answers. If the various stresses are compared it

is found that the shear stress is never in excess of sixty percent of the bending stress. Since the same approximations are applied to simplifying both stresses this is considered a worthwhile deduction.

2. Wave Characteristics Considerations

This section will simply outline the wave character which was assumed in the development of this paper. More exact methods or at least more complicated ones, of balancing a ship on a wave may be found in the references by Muckle and Murray listed at the end of this paper. These depend on various assumptions concerning wave heights and shapes and it can be shown, although this author only states it here, that variations of as much as thirty-three percent can occur in these methods depending on what shape wave (trochoidal, sine, etc.) or wave height ($L_w/20$, $1.1\sqrt{L}$) are assumed.

The assumptions made in this paper are:

a. The severest condition occurs when wavelength is equal to ship length.

b. The ship is assumed to be "at rest" on the wave, i.e., quasi-static.

c. Water pressure on the side is not related to the wave but is based rather on a maximum head of twice the draft. Pressure is often taken proportional to wave height which, based on $L_w/20$, is about eighteen feet in the example. Thus the double draft assumption is conservative.

d. The wave surface is a trochoid.

e. The wave height is taken as $L_w/20$.

3. Simplification of Failure Criteria

If in a Mohr's circle analysis longitudinal, transverse, and shear stresses are considered the following equations for the principle stresses may be devined by inspection:

$$\sigma_{max} = \frac{\sigma_x + \sigma_y}{2} + \left[\left(\frac{\sigma_x - \sigma_y}{2}\right) + \tau_{xy}^{2} \right]^{1/2}$$

$$\sigma_{min} = \frac{\sigma_x + \sigma_y}{2} - \left[\left(\frac{\sigma_x - \sigma_y}{2}\right)^{2} + \tau_{xy}^{2} \right]^{1/2} .$$

Then if

$$A = \frac{\sigma_x + \sigma_y}{2}$$

and

$$B = \left[\left(\frac{\sigma_x - \sigma_y}{2}\right)^{2} + \tau_{xy}^{2} \right]^{1/2}$$

it is seen that for

a. The Mises - Henke, Shear Strain Energy, Criteria

$$\sigma_{max}^{2} + \sigma_{min}^{2} - \sigma_{max}\,\sigma_{min} = \sigma_y^{2}$$

or

$$A^2 + 3B^2 = \sigma_y^{2} ,$$

b. The Rankine, Principle Stress, Criteria

$$\sigma_{max} = \sigma_y ,$$

or

$$A + B = \sigma_y .$$

c. The Guest Tresca, Maximum Shear, Criteria

$$\sigma_{max} - \sigma_{min} = \sigma_y$$

or

$$2B = \sigma_y,$$

d. The St. Venant, Maximum Strain, Criteria

$$\sigma_{max} - m\,\sigma_{min} = \sigma_y$$

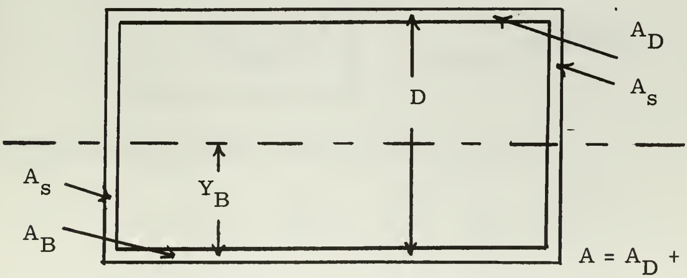
or

$$(1 - m)A + (1 + m)B = \sigma_y.$$

4. <u>Moment of Inertia</u>

This is the derivation of the Neutral Axis Location and Moment of Inertia equations which are set forth in step 4b of the design procedure. In addition, the format used in the derivation is considered useful for any moment of inertia calculations for a ship's section.

Consider for simplicity a rectangular ship's section



$$A = A_D + A_B + 2A_s$$

with areas of plating as noted.

Taking moments about the base of the section of the cross-sectional areas, we have, in tabular form,

Member	a	d	ad	ad^2	i_o
Deck	A_D	D	$A_D D$	$A_D D^2$	negligible
Sides	$2A_s$	D/2	$2A_s D/2$	$2A_s D^2/4$	$2A_s D^2/12$
Bottom	A_B	0	0	0	negligible

and $$\Sigma ad = \underline{\underline{D(A_D + A_s)}}$$

and $$\Sigma ad^2 + \Sigma i_o = \underline{\underline{D^2 \, (A_D + 2A_s/3)}}.$$

Then if

$$Y_B = \frac{\Sigma ad}{\Sigma a}$$

we have

$$\boxed{Y_B = \frac{D(A_D + A_s)}{A}} \qquad + \cancel{1/2 \, t_D} \text{ negligible}$$

and by the well-known transfer of axis theorem

$$I = D^2 \, (A_D + \frac{2A_s}{3}) - A \left[\frac{D^2 \, (A_D + A_s)^2}{A^2} \right]$$

or simplifying

$$I = \frac{D^2}{A} \left(AA_D + \frac{2}{3} AA_s - A_D^2 - A_s^2 - 2A_DA_s \right)$$

and finally

$$\boxed{I = \frac{D^2}{A} \left[A_DA_B + 2A_s \left(\frac{A}{3} - \frac{A_s}{2} \right) \right]} \quad .$$

APPENDIX B

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SNAME - Society of Naval Architects and Marine Engineers.

NECIES - North-East Coast Institution of Engineers and Shipbuilders.

TRINA - Transactions of Royal Institute of Naval Architects.

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